POWER

Volume 53 New York, May 17, 1921 *Number* 20

"EXPLAINING THE INVENTION THAT WILL MAKE THEM RICH"

New Power Plant of Monsanto Chemical Works

A modern power plant in which compound steam engines generate direct current for electrolytic work and discharge their exhaust to mixed-pressure turbo-generators which furnish alternating current for lighting and power. As the two loads and the water rates practically balance, this arrangement of equipment shows exceptional economy. Other services required of the plant are the supplying of compressed air, high-pressure steam and water to the chemical works.

FOR years the Monsanto Chemical Works has had in East St. Louis a large plant for the manufacture of heavy acids and chemicals. Electric power was purchased, but steam for process work and for heating was generated in the company's own boiler room. In 1919 the erection of a new plant to make chlorine gas, caustic soda, chlora benzol and sulphur chloride was begun and is still under way. As this new gas plant would require considerable steam at high pressure for process work, direct current to make chlorine gas electrolytically and alternating current for power and lighting, it was decided that these various services could be supplied more economically from a power plant erected upon the premises. As the boiler plant at the older works had become antiquated, the decision was made to abandon it and also to discontinue the purchase of current, the electrical and steam services for the two works both to be supplied from the new power plant.

In the electrolytic process of making chlorine gas direct current would be required, and the estimated load was 600 kw. This load would be continuous 24 hours a day, and any interruption would result in serious losses in the manufacturing process. Efficiency and continuity of service were the two factors determining the choice of prime mover, and of the two reliability came first. Three heavy tandem-compound engines, 16 and 30 by 30 in., driving at 150 r.p.m., 400-kw. 240-volt shunt-wound generators, were installed. Two of them must operate continuously to carry the load, with the third unit in reserve, to furnish 5½ tons of chlorine and 8 tons of the byproduct, caustic soda, per day of 24 hours. The load on each machine will range from 280 to 350 kw., depending on the deterioration of the graphite electrodes in the chlorine cells. It is thus evident that the generating units are somewhat larger than required for the present load, so that the heavy, slow-speed units

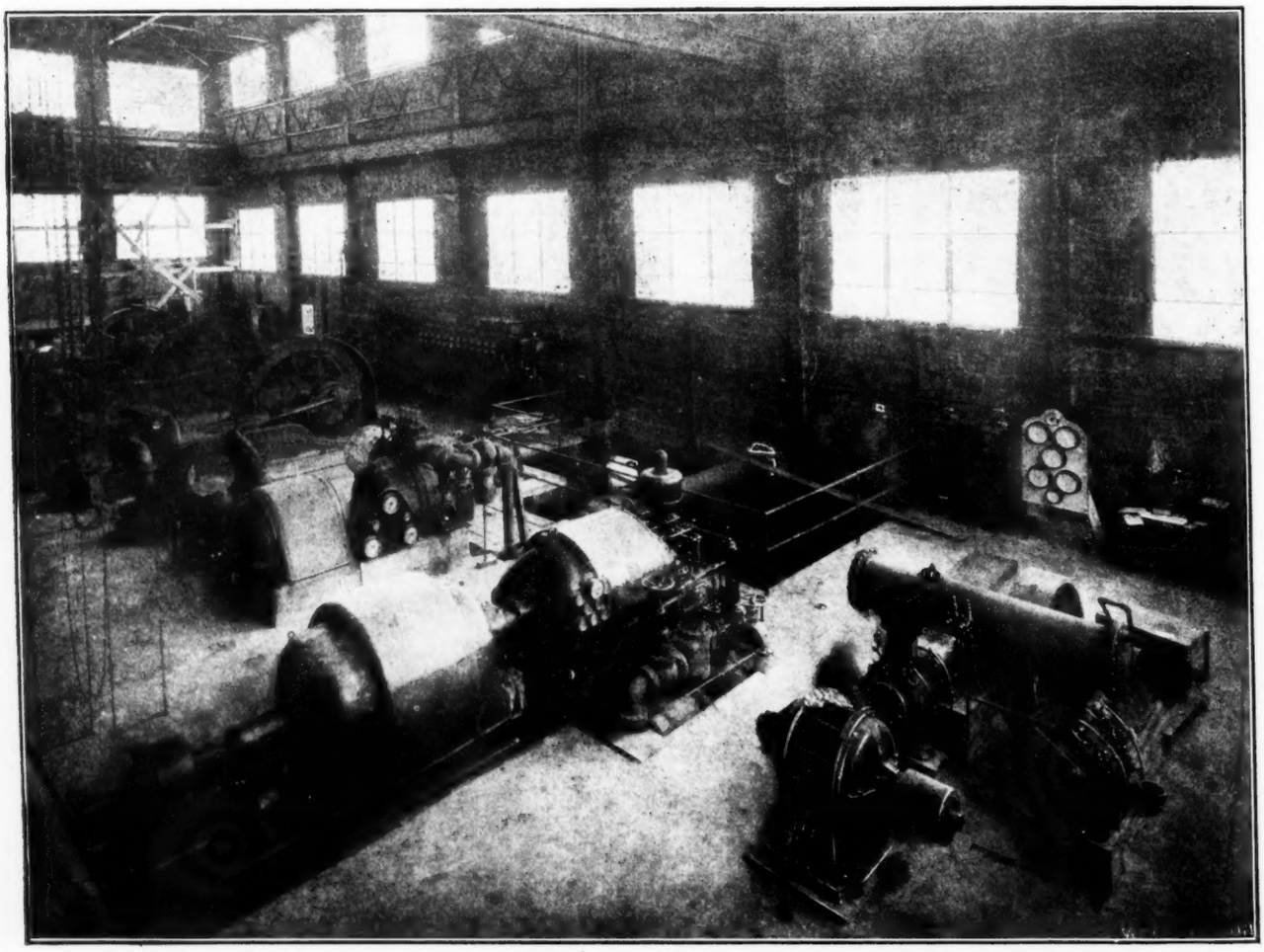

FIG. 1. VIEW OF THE GENERATING ROOM OF THE POWER PLANT OF THE MONSANTO CHEMICAL WORKS

under easy operating conditions should give reliable service for years, with maintenance reduced to a minimum. In addition a reserve machine increases the reliability of the service, and the three units all of the same size give a flexibility that could not be obtained with one or two machines of larger capacity.

Owing to the fact that the older plant had alternating-current equipment and that the distances for the transmission of electrical energy to the remote parts of the works were considerable, the continued use of alternating current for power and lighting was considered desirable, particularly as it would permit the installation of compact high-speed units. Mixed-pressure turbogenerators were the choice of the designing engineers, these units taking the exhaust steam from the compound engines, of which there would be a continuous supply in sufficient quantity to fulfill the requirements of the power and lighting load.

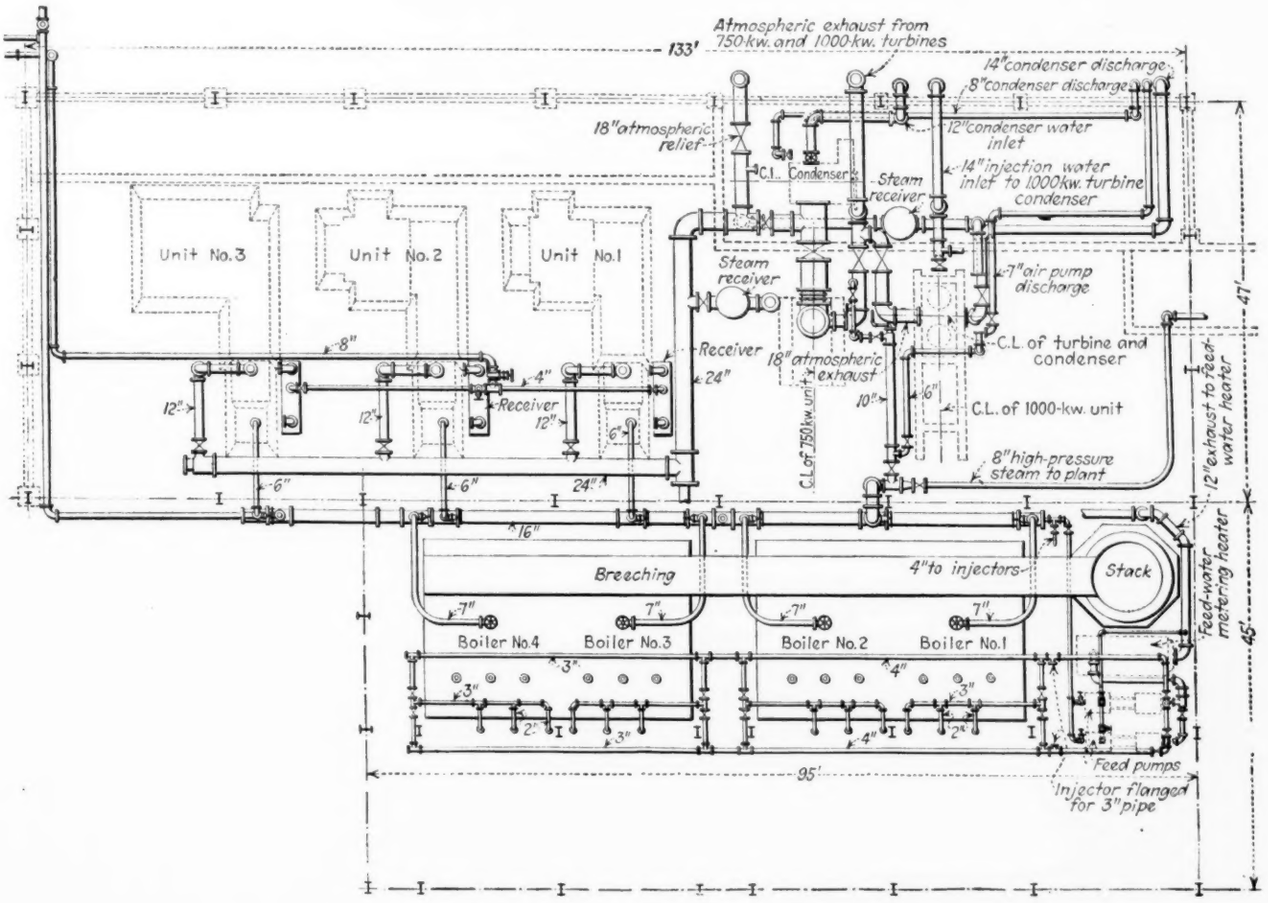

FIG. 2. PLAN OF STATION SHOWING LAYOUT OF PIPING

At present an alternating-current load of approximately 500 kw. is anticipated and from a brief interval of operation, a power factor of 92 per cent. This high power factor has been made possible by the judicious use of synchronous motors, and by the installation of another motor of this type of 150-hp. rating to drive an air compressor 24 hours a day, it is expected to bring the power factor to unity and practically to balance the direct-current load.

At 2-lb. back pressure the engines have a water rate of 35 lb. per kilowatt-hour and the mixed pressure turbines take 34 lb. of exhaust steam per kilowatt-hour. The direct-current load is continuous and practically constant, and the alternating current load is fairly uniform. To utilize whatever surplus exhaust steam may be available, a 4-in. line has been tapped to the intermediate receivers of the reciprocating units, taking steam at the prevailing pressure of 40 lb. for the evaporation of caustic liquor. The remainder of the steam required for this purpose is drawn from the boilers, passing through two pressure-regulating valves in being reduced from the boiler pressure of 175 lb. to the service pressure of 40 lb. The combination is thus an ideal one and should result in highly economical operation.

Two mixed-pressure units were installed, one of 750-kw. capacity and the other rated at 1,000 kw. The speeds are 3,600 r.p.m. in each case, and three-phase currents are generated at 440 volts. These machines are larger than the present load demands, but enlargement of the manufacturing plant will call for more alternating-current power, and there is capacity to meet a considerable increase. If necessary, live steam can be used to supplement the supply of exhaust steam to the mixed-pressure turbines, but it is quite probable that any enlargement of the plant will increase also the direct-current load and add to the supply of exhaust steam.

Jet condensers serve the turbines. The circulating pump is motor-driven in one case, and a turbine is the drive for the other pump. Condensing water is supplied from two deep wells at a rate of 1,350 gal. per

min. from each and a temperature of 55 deg. the year around. This water is also used in the works and for fire purposes, but is so heavy in minerals that it cannot be used in surface condensers or in the boilers. For the latter purposes water is drawn from the city mains, passing first through double-pipe condensers in the refrigerating plant for liquefying chlorine gas, then through the condensers in the chlora benzol building, and finally to the feed-water heater in the boiler room at a temperature approximating 110 deg. F.

From the power plant the outgoing electric feeders, steam and return lines, air lines and the water supply to the older plant are conducted underneath the street in a concrete tunnel 8 ft. wide and 7½ ft. high. The cables are suspended from the ceiling, and the steam and air lines rest on standard pipe supports. At the distribution house on the opposite side of the street from the plant, where the control valves and switches are located, the steam and air pipes and the electric cables are brought to the surface and for the rest of the way to the older works are carried overhead on steel trestlework. The water pipes continue underground.

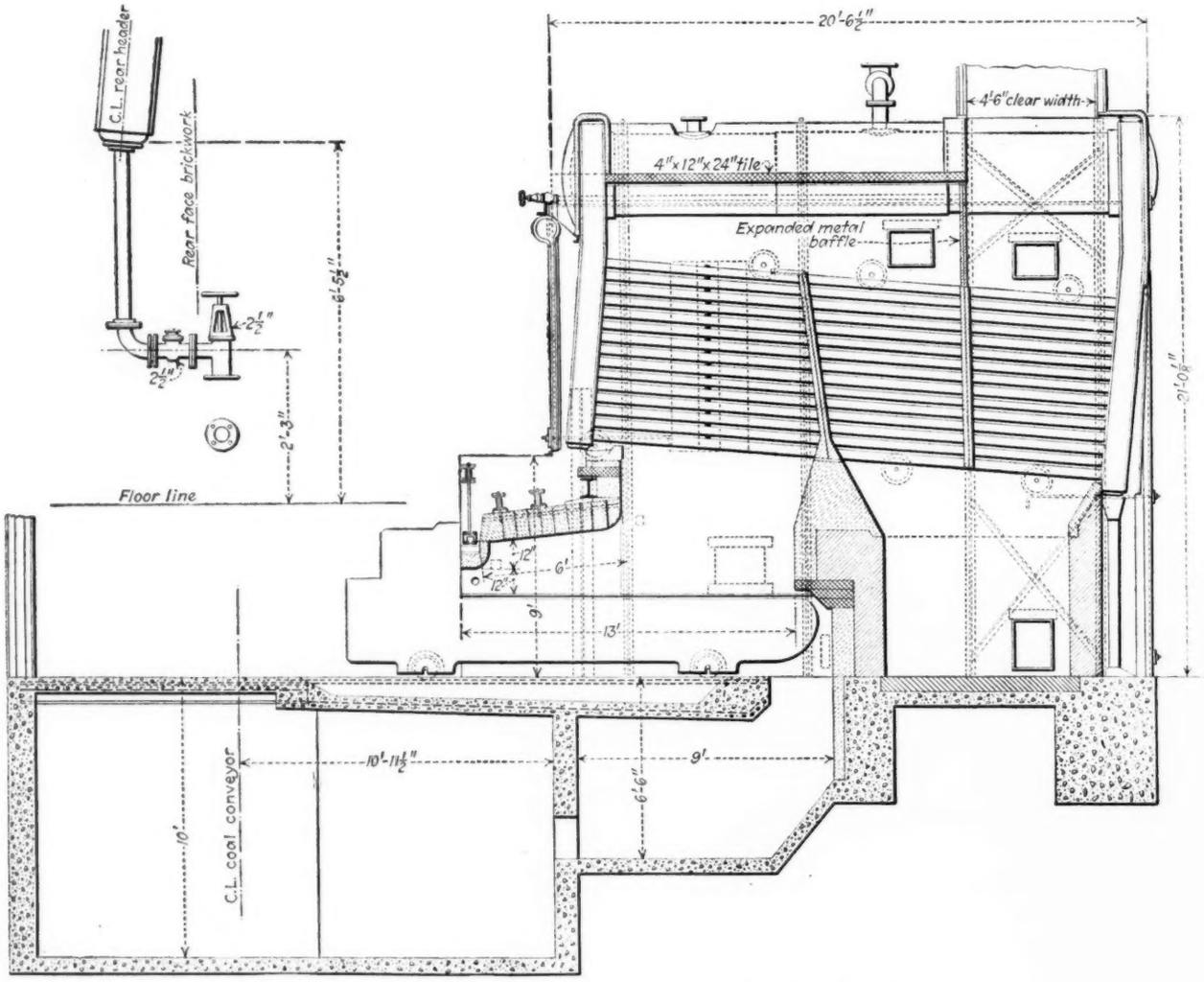

FIG. 3. SECTIONAL ELEVATION THROUGH BOILER SETTING

For excitation of the turbo-generators there is an exciter mounted on the shaft of the 1,000-kw. machine. The smaller unit has a turbine-driven exciter unit, and if need be, a 175-ampere-hour storage battery may be drawn upon temporarily. The usual functions of the battery are to operate the remote-controlled oil switches and the motor-operated valve on the turbine exhaust line.

Other equipment in the engine room are the air compressors, which supply air compressed to 100 lb. to the works for the agitation of acids during manufacture and a variety of other purposes. At present there are five machines, three steam-driven having an aggregate capacity of 950 cu.ft. per min., and two motor-driven compressors, one rated at 700 and the other at 1,100 cu.ft. per minute.

The engine room is 133 ft. long, 47 ft. wide and 38 ft. high to the bottom chord of the roof truss. An abundance of window area gives plenty of light and ventilation, and these advantages are enhanced by a monitor on the roof taking up one-quarter of the width of the room. The boiler room is 93 ft. long and 45 ft. wide, with a monitor over each battery of boilers. The building frame is of steel with hollow-tile curtain walls which will be covered by a cement gun inside and out. Construction of this character has been favored throughout the works.

The present equipment in the boiler room consists of four 6,000-sq.ft. boilers set two to the battery. They

are designed for a pressure of 200 lb., but operate at 175 lb. The plant has been designed with a view to increasing it to three times its present capacity without interfering with the equipment now installed. The next move is to be the installation of two more boilers of the size previously indicated.

Each boiler is served by a chain-grate stoker having 142 sq.ft. of active grate area, which bears a ratio to the steam-making surface of 1 to 42. Stokers of this comparatively large size were considered necessary to burn a sufficient quantity of the inferior coal available in the vicinity of St. Louis to maintain high capacities. A 6-ft. ignition arch 24 in. above the grate at the front and rising 30 in. above at the rear, has been installed. The support and arrangement of this arch is shown in the sectional view of the boiler, Fig. 3. From the bottom of the front tube header the boiler has been set 9 ft. above the floor. The furnace volume approximates 540 cu.ft., or about 3.8 cu.ft. per square foot of grate, which, on the basis of 30 lb. of coal per square foot, would give 0.127 cu.ft. of furnace volume per pound of coal burned. The tubes are baffled vertically for three passes, with a built-up bridge wall and a baffle sloping toward the front confining the first pass. At the rear the gases rise vertically to the outlet passing around the drums. They enter an unlined steel breeching of uniform section, which has been built in a straight line over the uptakes to the stack. This breeching has a cross-sectional area of 63 sq.ft. and discharges to a self-supporting

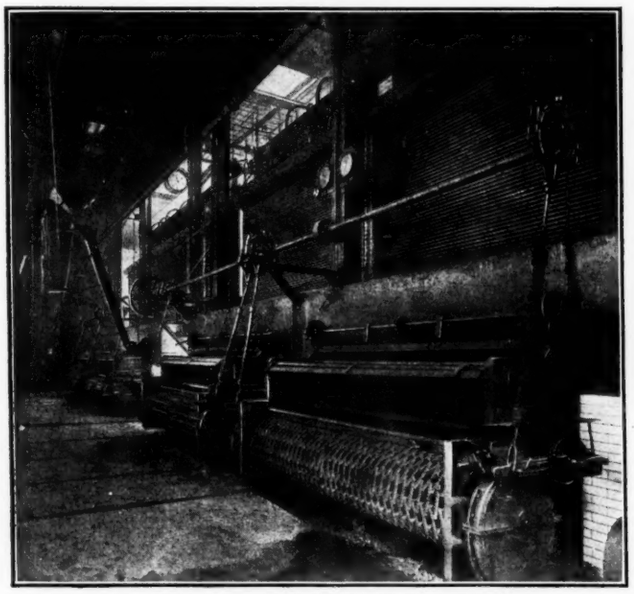

FIG. 4. VIEW OF BOILER FRONTS

steel stack having a diameter at the top of 8 ft. and an area of 50 sq.ft.

Reviewing the boiler dimension, per 100 sq.ft. of steam-making surface there have been provided 2.4 sq.ft. of grate, 9 cu.ft. of furnace volume, 0.26 sq.ft. of smoke flue and 0.21 sq.ft. of stack area. Per square foot of grate 3.75 cu.ft. of furnace volume has been provided.

The stack rises 163 ft. above the grate. It weighs 64,706 lb. and is supported on a concrete foundation having a total weight of 585,827 lb. Thus the aggregate weight of stack and foundation is 650,533 lb. As the soil is chiefly sand in this vicinity, having originally formed the bed of the river, a large bearing area was required to support the weight of the stack. The concrete foundation has been stepped out from 12 ft. diameter at the floor line to an octagonal base 21 ft. between sides. This gives a base area of 363 sq.ft. and a bearing pressure of 1,792 lb. per sq.ft. As indicated in Fig. 5, that portion of the foundation above

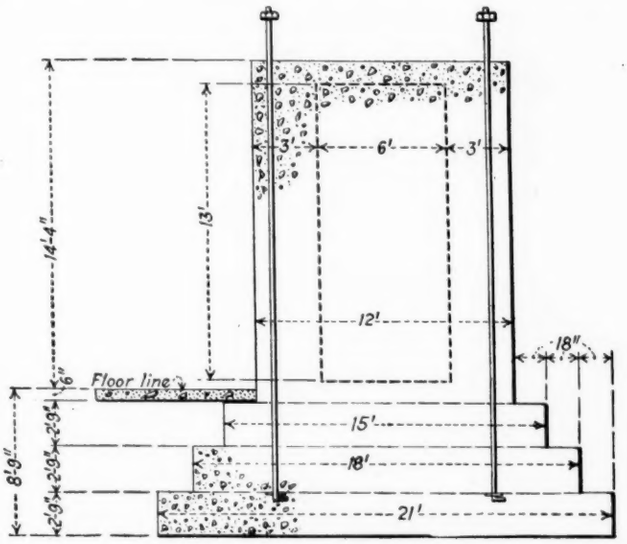

FIG. 5. CONCRETE FOUNDATION OF SELF-SUPPORTING STEEL STACK

the floor line has been made hollow, the diameter of the hole being 6 ft. and the depth 13 ft., leaving a concrete shell on either side 3 ft. thick. This was done to reduce the weight and incidentally to lower the center of gravity by 8 in. Against a wind pressure of 100 miles per hour the stack has been designed to give a factor of safety of 2.5. It is secured to the foundation by sixteen 2½-in. anchor bolts which tie in with the steel reinforcing at the bottom of the foundation.

At the rear each stoker has four dampers to shut off 10, 20, 30 and 40 per cent of the grate area. Thus on light loads the grate surface can be reduced by steps of 10 per cent to 60 per cent of the original area, so that the fires may be maintained active and in proper condition to give high combustion efficiency. The drive is by means of eccentrics from an overhead lineshaft, which in turn is belted to a jack shaft having a silent-chain-drive connection to a motor mounted on one of the building columns. Normally, this motor drives the stokers, but upon occasion it is relieved by an auxiliary drive consisting of a vertical engine set on the boiler-room floor. This arrangement provides for any stoker drive emergency that may arise.

An open metering heater having a capacity of 170,000 lb. of water per hour is a part of the boiler-room equipment. It is mounted on a foundation to raise it above the boiler-feed pumps, which are located on the boiler-room floor. There are two of these pumps, each of the direct-acting duplex type and having capacity to serve six boilers, while the other is held in reserve. As indicated in Fig. 2, a double feed line has been provided with a cross-connection at each boiler through which the feed water may pass from either main into the line supplying the boiler. As a safeguard a 3-in. injector has been placed in either feed line, so that if both pumps should fail, there would still be a means

of feeding water to the boilers. There are feed-water regulators to control the supply and governors on the steam pumps.

Coal for the plant is delivered by rail. From a track hopper it is transferred by an apron conveyor to a continuous bucket conveyor, passing over an overhead steel bunker of parabolic type. The bunker has a capacity of 450 tons, which is about a three-days' supply. Illinois screenings containing 11,500 B.t.u. per pound is the fuel used. To protect the steel shell the bunker is lined with 1-in. shale tile set in bitumastic cement. This lining is impervious to the acids drained from wet coal.

Opposite each boiler double-cut gates in the bottom of the bunker discharge the coal to a weighing larry of 1,500 lb. capacity. The latter is of the traveling type with provision to chute the coal to the stokers after the weight has been ascertained. The coal-handling system has a capacity of 40 tons per hour, and the conveyors employed are motor-driven.

To provide for cooling the coal in case it should heat in the overhead bunker, square openings have been made in the boiler-room floor directly under the coal gates in the bunker. By opening these gates, the coal drops through the holes in the boiler-room floor into the continuous bucket conveyor underneath, which returns it to the bunker. The time interval is sufficient to cool off the coal and prevent it from heating to the danger point.

Under each stoker an ash hopper has been provided of sufficient capacity to give the ashes a chance to cool before they are transferred by the continuous bucket conveyor to a partitioned-off portion of the steel bunker. Portable chutes lead from the door of the ash hopper to the continuous bucket conveyor, so that in one operation the ashes may be raked from hopper to conveyor. From the overhead ash bunkers, steel chutes lead out over the railway track. These chutes are of the counterbalanced, hinged type, so that they may be easily raised or lowered when in and out of use.

Referring again to Fig. 2, it will be easy to trace the high-pressure steam and the exhaust piping. Seven-inch leaders connect each boiler to the 16-in. extra-heavy main steam header supported by the partition wall. Provision has been made for a continuation of the header when the addition to the boiler room, previously referred to, is made. All this high-pressure piping is of steel with Van Stone joints and copper gaskets. Long-radius bends care for the pipe expansion.

The low-pressure piping shows the connection to the intermediate receivers of the engine units and the 8-in. line through which the steam at 40 lb. pressure is conducted to the chemical plant. Each engine exhausts to a 24-in. main common to all, from which the steam passes on to the low-pressure turbines or escapes to atmosphere should there be a surplus supply. All high-pressure piping has 2½ in. of nonpareil covering. The low-pressure piping has the same kind of covering. On pipes above 8 in. in diameter it is 2 in. thick, and 1 in. thick on the smaller piping.

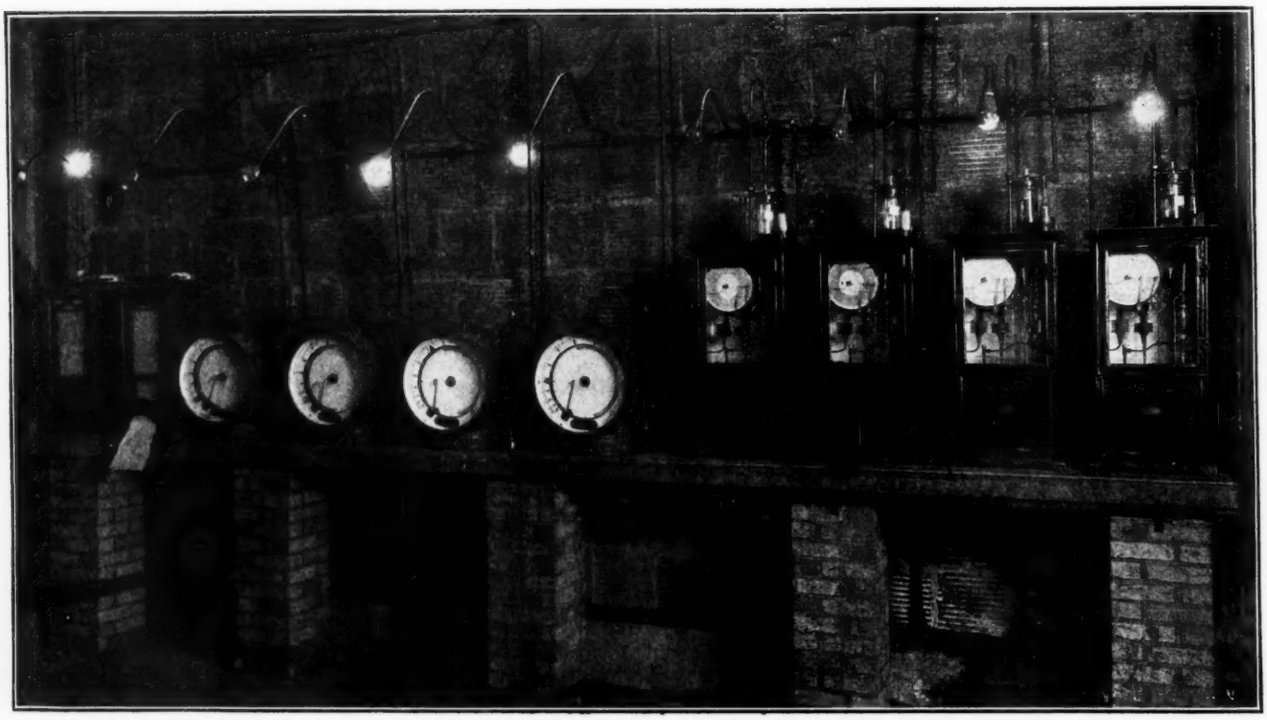

FIG. 6. BOILER INSTRUMENTS CONVENIENTLY LOCATED

Before closing the description of the plant, it will be of interest to enumerate the unusually complete complement of instruments provided to give an accurate record of all essential operating data. Each boiler is equipped with a differential draft gage. There are two multiple recording pyrometers to give the temperature of the flue gases from the four boilers, a CO_2 recorder and a flow meter on each unit, and in addition, flow meters on all steam lines going out of the plant.

There are also flow-meters on the high-pressure lines leading to each turbine to record the makeup

MONSANTO CHEMICAL WORKS

POWER PLANT LOG **EAST ST. LOUIS PLANT**

DATE..... 192 .

Time	Engine No. 1 Volts	Engine No. 1 Amp.	Engine No. 2 Volts	Engine No. 2 Amp.	Engine No. 3 Volts	Engine No. 3 Amp.	Turbine No. 1 Volts	Turbine No. 1 Amp.	Turbine No. 1 Watts	Turbine No. 1 Vacuum	Turbine No. 1 Oil Pressure	Turbine No. 1 Discharge Water Temp.	Turbine No. 2 Volts	Turbine No. 2 Amp.	Turbine No. 2 Watts	Turbine No. 2 Vacuum	Turbine No. 2 Oil Pressure	Turbine No. 2 Discharge Water Temp.	Line Water Temp.	Power Factor	Barometer Reading	Steam Pressure High Pressure Line	Steam Pressure Engine Receiver	Steam Pressure Low Pressure Exhaust
1 A.M.																								
2																								
3																								
4																								
5																								
6																								
7																								
8																								
9																								
10																								
11																								
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1 P.M.																								
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Total K.W.H. Turbine No. 1		Current K.W.H. Circuit No. 1	
Total K.W.H. Turbine No. 2		Current K.W.H. Circuit No. 2	
Total K.W.H. Turbine No. 3		Current K.W.H. Circuit No. 3	
Total K.W.H. Turbine No. 4		Current K.W.H. Circuit No. 4	
Steam Consumed, Turb. No. 1	Lbs.	Current K.W.H. Circuit No. 5	
High Pressure	Lbs.	Current K.W.H. Circuit No. 6	
Low Pressure	Lbs.	Current K.W.H. Circuit No. 7	
Steam Consumed, Turb. No. 2	Lbs.	Current K.W.H. Circuit No. 8	
High Pressure	Lbs.		
Low Pressure	Lbs.		
Steam Consumed, Turb. No. 3	Lbs.		
High Pressure	Lbs.		
Low Pressure	Lbs.	Turbine Oil Used	Gals.
H.P. Steam Cons'd, Plant "B"		Engine Oil Used	Gals.
H.P. Steam Cons'd, Plant "C"		Compressor Oil Used	Gals.
L.P. Steam Cons'd, Plant "C"		Cylinder Oil Used, No. 1	Gals.
H.P. Steam Cons'd, Plant "D"		Cylinder Oil Used, No. 2	Gals.
		Steam Output, Boiler No. 1	Lbs.
		Steam Output, Boiler No. 2	Lbs.
		Steam Output, Boiler No. 3	Lbs.
D.C. K.W.H. No. 1		Steam Output, Boiler No. 4	Lbs.
D.C. K.W.H. No. 2		Feed Water Used	
D.C. K.W.H. No. 3		Coal Consumed	
		Coal Received	Lbs.
		Coal on Bunker	

	From	To
ENGINEERS		
ON	From	To
WATCH	From	To

Air Compressor No. 1—Started Stopped

Air Compressor No. 2—Started Stopped

Air Compressor No. 3—Started Stopped

Air Compressor No. 4—Started Stopped

REMARKS.

Signed:

FIG. 7. SAMPLE OF POWER-PLANT LOG

steam that may be necessary, and a flow meter has been connected to the lines tapping the engine receivers to give a record of the steam drawn from this source for process work. There are recording pressure gages on the high-pressure steam lines to the engines and on the low-pressure lines to the turbines. Recording thermometers give the temperature of the condenser water, and a recording gage shows the vacuum that is carried.

A recording meter measures the water fed to the boilers, and its record serves as a check on the flow-meter readings. There is also a recording thermometer on the feed water, and as previously stated, the coal is weighed, so that all operating data that could be desired on the steam end of the station are immediately available. On the electrical side every circuit to the various departments in the two works is metered as well as the output of the generating units. In addition all instruments essential to operation have been provided, so that the engineer of such a plant should have no difficulty in determining the cost of his products, steam and electrical energy, and the amount of each that is used in the power plant itself and by each department of the two works. Fig. 7 shows the daily power-plant log on which the records are kept.

For the design and erection of the plant credit is due D. Larkin, of Larkin & Pratt, consulting engineers, of St. Louis.

PRINCIPAL EQUIPMENT OF POWER PLANT OF MONSANTO CHEMICAL WORKS

No.	Equipment	Kind	Size	Use	Operating Conditions	Maker
4	Boilers	Hot. water-tube	6,000 sq.ft	Generate steam	175 lb., natural draft stokers	Edge Moor Iron Co.
4	Stokers	Chain-grate	143 sq.ft	Serve boilers	Motor and engine drive	Illinois Stoker Co.
1	Weigh larry	Traveling	1,500 lb	Weigh coal to boilers		Stephens-Adamson Co.
1	Conveyor	Apron	40 tons per hr	Ry. track hopper to bucket conv.	Driven by 5-hp. G.E. motor	Stephens-Adamson Co.
1	Conveyor	Cont.-bucket	40 tons per hr	Elevate coal to bunker	Driven by 15-hp. G.E. motor	Stephens-Adamson Co.
1	Bunker	Parabolic	450 tons	Overhead storage	Steel lined with shale tile in bitumastic cement	Stephens-Adamson Co.
1	Heater	Open	170,000 lb. per hr	Heat feed water	Exhaust-steam	Harrison Safety Boiler Works
1	Pump	Duplex	14x10x12-in	Boiler feed		Worthington Pump & Machinery Co.
1	Pump	Duplex	12x7x10-in	Boiler feed		Worthington Pump & Machinery Co.
3	Engines	Compound	16x30x30-in	Generate direct current	175 lb. steam, 150 r.p.m.	McIntosh & Seymour
3	Generators	Shunt-wound	400-kw	Generate direct current	240 volts, 150 r.p.m.	Westinghouse Electric & Mfg. Co.
1	Turbine	Mixed-pressure	1,000-kw	Generate alternating current	Use exhaust from engines, 3,600 r.p.m.	Westinghouse Electric & Mfg. Co.
1	Generator	A. C.	1,000-kv	Generate alternating current	3-phase, 60-cycle, 440 volts	Westinghouse Electric & Mfg. Co.
1	Turbine	Mixed-pressure	750-kw	Generate alternating current	Use Exhaust Steam from engines, 3,600 r.p.m.	General Electric Co.
2	Condensers	Jet		Serve turbines	Motor- and turbine-driven circ. pumps	Westinghouse Electric & Mfg. Co.
1	Exciter		10-kw	Serve 750-kw. turbine	Driven by Terry turbine	General Electric Co.
1	Storage battery	M. V.	175-amp.-hr	Remote-control		Exide Storage Battery Co.
1	Crane	Conkey hand-operated	15-ton	Serve engine room	Span 45 ft	
1	Air compressor	Steam-driven	250 cu.ft. per min	Comp. air to works	100 lb. air	Ingersoll-Rand Co.
2	Air compressor	Steam-driven	350 cu.ft. per min	Comp. air to works	100 lb. air	Laidlaw-Dunn-Gordon Co.
1	Air compressor	Motor-driven	700 cu.ft. per min	Comp. air to works	100 lb. air	Chicago Pneumatic Tool Co.
1	Air compressor	Motor-driven	1,100 cu.ft. per min	Comp. air to works	100 lb. air	Laidlaw-Dunn-Gordon Co.

Lane & Bowler deep-well system; Richardson-Phenix oiling system; Rochester force-feed lubricators; Copes feed-water regulators; Defender pump governors; Diamond soot blowers; Schutte & Koerting injectors; Detroit steam traps; G. E. steam flow meters; Cochrane feed-water meter; Hays CO_2 recorders; Brown pyrometers; Foxboro recording thermometers and pressure gages; Defender dif.-draft gages; Crane and Chapman steam and water valves; Homestead and Faber blowoff valves; Link-Belt silent-chain drive on stoker motor; General Electric switchboard.

Testing a Four-Thousand Kilowatt Generator for Ground

By L. R. Gray

Lightning recently struck the line while a 4,000-kw. 2,300-volt generator was in operation, and broke down the insulation in several places. When the accident happened, considerable smoke was produced, showing that the insulation was badly burned. The machine was stopped and the end bells removed, when it was seen that one of the terminals was fused through and the insulation of the stator burned off in several places. The terminal was welded together with an acetylene-welding outfit and the damaged insulation was taped up.

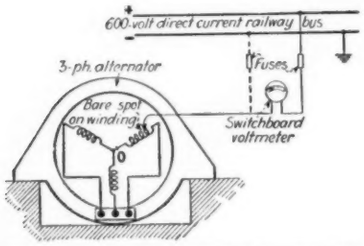


FIG. 1. GROUND TEST WITH 600-VOLT DIRECT CURRENT

It was then necessary to test the insulation for grounds to make sure that it was sound. One terminal of a magneto-and-bell testing outfit was first attached to the base of the machine and the other touched to a bare place on the stator winding; the bell rang, showing a ground. It was then decided to repeat the test with 600-volt direct current from the trolley circuit. To do this, the negative lead of the switchboard voltmeter on the direct-current panel was disconnected from the bus and connected to the winding, as in Fig. 1. Since it was a railway circuit, the other side of the circuit was already grounded and it was not necessary to run another wire. When the circuit was closed by inserting the fuse, the voltmeter should have shown a deflection if there was a ground, due to the current flowing, but to our surprise the needle did not move from zero. The magneto had already shown a ground, and this test showed that there was none; so, to decide these conflicting indications, it was decided to test the winding with the voltage for which it was guaranteed.

The rated voltage was 2,300; so the insulation should stand, by the A. I. E. E. Standards, $2 \times 2{,}300 + 1{,}000 = 5{,}600$ volts. However, 4,600-volt current was used, as it was more convenient to obtain and high enough for the purpose. Two ordinary 220-volt/2,300-volt distributing transformers were connected as shown in Fig. 2, with their primaries in parallel and their secondaries in series. The fuse block was fused with 15-ampere wire. One terminal was grounded on the machine frame. At the other terminal the bare end of the wire was taped to a stick of dry wood held by a man on an insulated stool. After the switch was closed, this man touched the wire to the bare place on the winding and drew an arc. If the arc was heavy and sustained, it would indicate a ground. When he first touched the winding, the 15-ampere fuse was blown, and upon repetition a 30-ampere fuse was also blown, showing a ground without question. The next thing was to locate the failure and repair it.

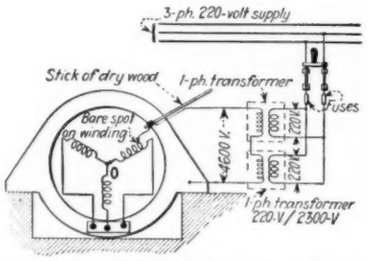


FIG. 2. GROUND TEST WITH 4,600-VOLT ALTERNATING CURRENT

First, the cables in the conduits were disconnected from the armature terminals and each one tested separately by the method shown in Fig. 2. The cables tested clear and the armature still showed a ground. Examination showed that the machine was star-wound with 36 coils. The winding was opened at the point *O*, Fig. 2, where one phase joined the neutral; this left 12 coils in one section and 24 in another. The two were tested separately and the ground found in the smaller section. Opening this in the middle left six coils in each section which were again tested separately. Opening up the grounded section at the middle left three coils in each section and so on; in this way the trouble was narrowed down by successive tests until the bad coil had been located.

Repairing the coil depends entirely on the machine and the break, so the method will not be described here. In this case we had no spare coils, and the insulation had to be repaired with mica and fishpaper.

After the repair was finished and the coil put in place, the winding was again tested and found free from grounds. A second test, however, blew the fuse and a third one drew an arc. Another ground was located and found to be in the section that had previously tested clear. Our testing methods were apparently breaking down the insulation that was in good condition before it was tested, although the voltage was not too high. It seemed probable that the arc drawn by the test set up high-frequency currents which broke down the insulation and caused the new grounds, and it was suggested that a water rheostat be used to make and break the circuit and thus damp out the harmful oscillations. To make the water rheostat a five-gallon bucket was filled with water, and a double handful of salt added. Then one end of the 220-volt wire was connected to a piece of iron plate that was dropped in the bottom of the bucket, the other end being handled by means of a stick of wood. The method of testing was then to make the connection on the winding of the machine and close the circuit through the water rheostat; if there was a ground, the fuse would blow. The resistance of the water damped the high-frequency currents or prevented their being set up, by limiting the first rush of charging current into the windings, thus saving the insulation. This use of a water rheostat as a switch is recommended in similar cases, as it ended our troubles at once.

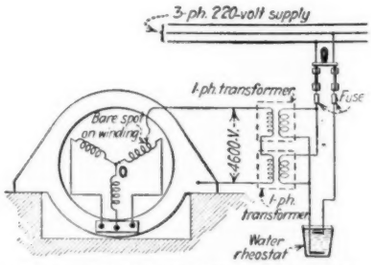


FIG. 2. GROUND TEST WITH 4,600-STAT TO ELIMINATE HIGH-FREQUENCY OSCILLATIONS

[We heartily indorse the author's recommendation. Such a precaution usually receives little thought in insulation tests, yet it is probable that the actual maximum voltage stress that the insulation must withstand in the usual tests is materially above the applied voltage. Under special conditions it may even reach many times the impressed voltage.—Editor.]

Draft and CO_2 Influence on Evaporation

Relation Between Combustion Rate and Draft—Derivation of C. R. Formula—An Automatic Stoker Speed and Draft Indicator

By JAMES T. BEARD, JR.

THE relation between furnace draft and combustion rate of coal is one of the least understood combustion problems. The idea is prevalent that an increase in draft will add to the rate of burning the coal, and, therefore, the rate of steaming, or horsepower, of the boiler. This idea is all right as far as it goes, but it is only half of the answer.

It is not surprising that the fireman believes that if more steam is wanted all he needs to do is to speed up the fan. This statement is a little unfair to him because he also knows that as his fire burns down he must keep throwing in coal. The fact remains, however, that he does not realize that after he has reached the limit of his air capacity he can still further raise the evaporation rate by increasing the efficiency of burning the coal, as evidenced by the increased percentage of CO_2 in the gases of combustion. This statement does not apply so much to the central station, where a good CO_2 record is generally maintained at all ratings, but it assuredly does apply to the large majority of industrial plants, where the average record generally varies from eight to ten.

In discussing "Rate of Combustion" in "Steam" the Babcock & Wilcox Co. says: "The amount of coal which *can be burned* per hour per square foot of grate surface is governed by the character of the coal and the draft available." This statement is correct, but it must

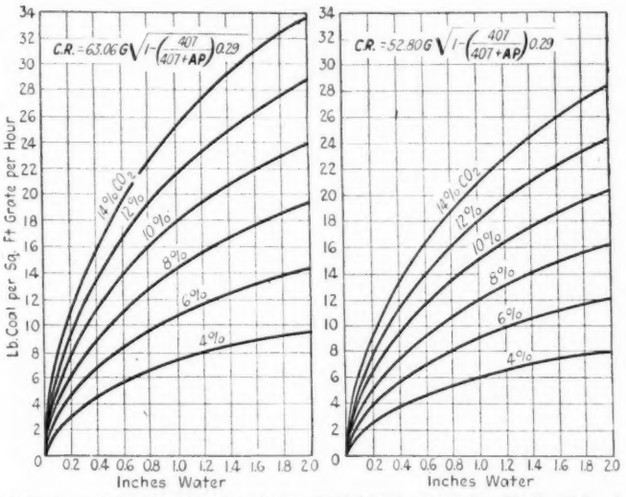


FIG. 1. COMBUSTION RATE CURVES (LEFT) FOR NO. 2 BUCKWHEAT AND (RIGHT) FOR NO. 3 BUCKWHEAT

be read as it is written; namely, "*can be burned*" and not "*is burned*." The amount of coal of a given character which is actually burned per square foot of grate per hour may or may not bear a definite relation to the draft available. If the percentage of CO_2 in the flue gases is the same in every case, it will. If not, there will be another story.

It is evident that many steam engineers have misread this and similar statements, else the misconception that available draft is the sole factor governing combustion rate would not be so general. I have met combustion engineers and technical men who have tried to plot values of combustion rates against pressure drops through fuel beds that had been obtained in actual practice. The chart as a rule looked like a lot of hen tracks. These men have usually concluded that there should be

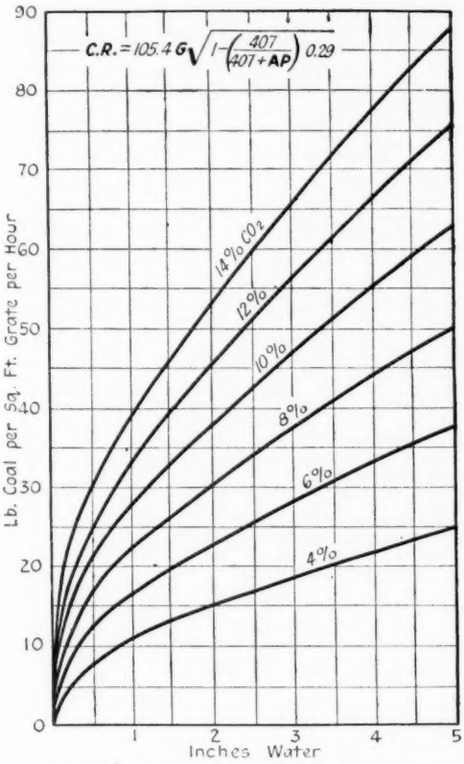


FIG. 2. COMBUSTION RATE CURVE FOR BITUMINOUS SLACK

some such relation in theory, but that it was an elusive thing to find in practice. And the whole trouble was that they did not take all the elements of the problem into consideration.

For example, no reasonable man will contend that the same combustion rate will be maintained with a new fire as with a fire half burned out. Granting that the coal and pressure drop through the fire bed is the same, the percentage of CO_2 in the gases will be greater in the first case than in the second. More coal is, therefore, being consumed in a given time, for where would the greater amount of carbon as CO_2 come from were it not from the coal?

In a given plant and with a given coal, therefore, the amount burned per square foot of grate surface per hour is governed by the pressure drop through the fuel bed and the percentage of CO_2 in the furnace gases. While it is interesting to know that both of these factors affect the combustion rate, it is more valuable to know just how much and in what way they do so. The writer has developed a general equation for combustion

rate in terms of pressure drop and percentage of CO_2. Although he hesitates to burden the readers of *Power* with a lot of mathematics, still the equation has been helpful to him and he thinks it of sufficient value to pass along to the next man.

The weight of air in pounds theoretically required to burn completely one pound of coal and leave no surplus oxygen is,

$$Lb.\ air\ per\ lb.\ coal = 11.58\ C + 34.8\left(H - \frac{O}{8}\right)$$

where C, H and O are respectively the percentages of total carbon, hydrogen and oxygen in coal by ultimate analysis. An average anthracite runs about 72.70 total carbon, 2.15 hydrogen and 1.88 oxygen. Substituting these values in the equation, complete combustion requires 9.08 lb. of air for every pound of coal. The theoretical maximum CO_2 obtained in such a reaction would be 19.85 per cent.

In practice, of course, a considerable excess of air is necessary in order to get enough air at the coal to burn

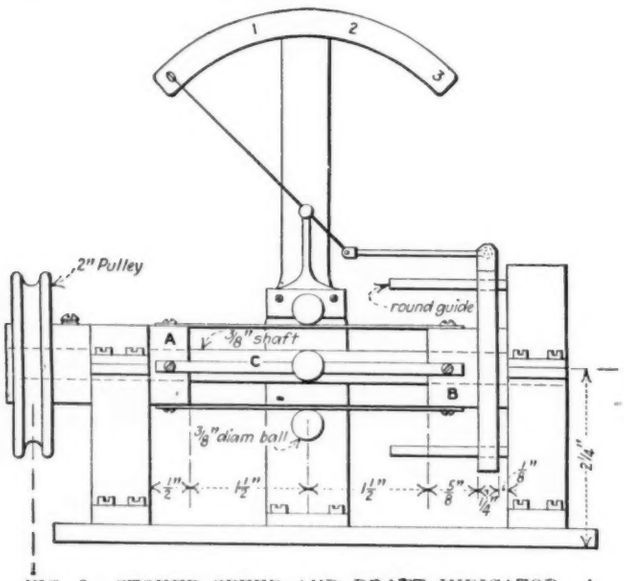


FIG. 3. STOKER SPEED AND DRAFT INDICATOR. *A*, COLLAR FIXED ON SHAFT; *B* SLIDING COLLAR; *C*, STEEL SPRINGS

it at all under the conditions obtaining in boiler plants. Since a deficiency of CO_2 usually denotes excess of air in which case CO_2 is in reverse ratio to the air actually used the percentage of excess air is represented by:

$$\frac{G_t - G_a}{G_a} \quad \text{or} \quad \frac{G_t}{G_a} - 1$$

where G_t is the theoretical maximum CO_2 (per cent) and G_a is the actual CO_2 obtained (per cent). The weight of air in pounds actually supplied to burn one pound of coal in practice is, therefore,

$$Lb.\ air\ per\ lb.\ coal = \left[\ 11.58\ C + 34.8\ \left(H - \frac{0}{8}\right)\right]\frac{G_t}{G_a}$$

or (for anthracite) substituting the value of 19.85 for the theoretical CO_2 and using 9.08 as the weight of air theoretically required, the air actually supplied becomes

$$\frac{180.24}{G_a}$$

Now,

$$Lb.\ coal\ per\ hr. = \frac{Lb.\ air\ per\ hr.}{Lb.\ air\ per\ lb.\ coal}$$

and therefore

$$C.\ R.\ (combustion\ rate) = lb.\ coal\ per\ sq.ft.\ grate\ per\ hr. = \frac{Lb.\ air\ per\ hour}{Lb.\ air\ per\ lb.\ coal \times Grate\ Area} = \frac{Lb.\ air\ per\ hr. \times G_a}{180.24 \times Grate\ Area}$$

Here the writer has taken liberties with the formula for the flow of air. Perhaps you will think that he has tampered with mathematics or been guilty of drawing upon his imagination, but the end justifies the means, and he has succeeded in obtaining an expression that is fairly accurate and of practical value.

The air is flowing under pressure or draft, as the case may be, through the spaces between the individual lumps of coal. The sum of the areas of all these spaces represents the area through which the entire volume of air must pass and bears a fairly constant relation to the grate area for a given size of coal. Thus,

$$Area\ air\ passage = K \times Grate\ Ar$$

The weight of air in pounds flowing per second is,

$Lb.\ air\ per\ second =$

$$109.1\ (K \times Grate\ Area)\ C\delta\sqrt{T_1\left[1-\left(\frac{P_2}{P_1}\right)^{.29}\right]}$$

where

C = Coefficient of discharge;
δ = Density of air at P_2 and T_2;
T_1 = Temperature air in boiler room (550 deg. abs.);
P_2 = Absolute pressure air in furnace;
P_1 = Absolute pressure air in boiler room.

Converting this expression into pounds of air per hour and substituting it in the previous equation for combustion rate,

$$C.R. = 50{,}853\ KCG\delta\sqrt{1-\left(\frac{P_2}{P_1}\right)^{.29}}$$

The calculated values of K and experimentally determined values of C for various sizes of coal are given as follows:

Coal	Value C	Value K
No. 1 buckwheat	0.311	0.392
No. 2 buckwheat	0.252	0.408
No. 3 buckwheat	0.204	0.422
Bituminous slack	0.432	0.423
Bituminous nut	0.646	0.363

For different coals the equation, therefore, becomes,

$$C.R. = MG\sqrt{1-\left(\frac{407}{407+P}\right)^{.2}}$$

where

G = percentage of CO_2 in furnace gases;
P = pressure drop in inches water through fuel bed;
M = constant = 74.76 for No. 1 buck, 63.06 for No. 2 buck, 52.80 for No. 3 buck, 105.40 for Bit. slack, 135.27 for Bit. nut.

This is the proper form of the equation for the combustion rate of coal, and it shows that it depends upon two variables—the percentage of CO_2 in the furnace gases and the pressure drop through the fuel bed. The constant M is determined by the character and size of the coal burned. It is evident from the expression that the combustion rate increases directly in proportion to an increase in the percentage of carbon dioxide in the gases, but that its relation to the drop through the fuel bed is somewhat more complex, increasing directly as the expression contained in the radical increases. Furthermore, a given increase in percentage of CO_2 will cause the combustion rate to rise more rapidly than will a corresponding increase in the pressure drop.

The importance of maintaining a high CO_2 record, not only for the sake of efficiency but also for the sake of capacity, is therefore obvious. For example, assume that the draft loss through the fuel bed is 0.8 in. of water and the furnace gases contain 8 per cent CO_2. For No. 3 buckwheat these conditions correspond to a combustion rate of 10.7 lb. of coal per square foot of grate per hour. By boosting the CO_2 to 12 per cent (a 50 per cent boost) and keeping the drop through the fuel bed the same, the combustion rate would be raised to 16 lb. On the other hand, if the CO_2 is allowed to remain the same, it would be necessary to increase the drop through the bed to nearly 2 in. (1.96 in.) of water in order to get the same combustion rate, an increase of 145 per cent in the air pressure. If the stack or the fans are not good for the increase, then the steam pressure drops. But whether the steam drops or not, any increase in combustion rate or capacity that is obtained by air pressure at the expense of CO_2 or to the neglect of CO_2 is expensive, to say the least.

The charts accompanying this article have been plotted from the equation and have been checked up from time to time in actual practice. They have been of assistance to the writer at times and he is passing them along, as they are more convenient for every-day use than the formula.

As to the practical value of the foregoing, let us assume that a plant is burning bituminous slack on a multiple-retort stoker with forced draft. The writer has a particular plant in mind where this information was of considerable help. The automatic regulators were disconnected from the fan engines and were hooked up to operate on the stoker engines. When the steam pressure dropped the stokers speeded up and the coal fed faster. The writer then designed and had built the indicating device shown in the cut. This was belted to the stoker-engine shaft and was at first calibrated to read in speed of stoker engine, but later in pounds of coal per square foot of grate per hour. Then, by referring to the chart for bituminous coal, the scale was again converted into inches of water drop through the fuel bed. Thus for all stoker speeds the indicator showed the proper drop through the fuel bed for good combustion.

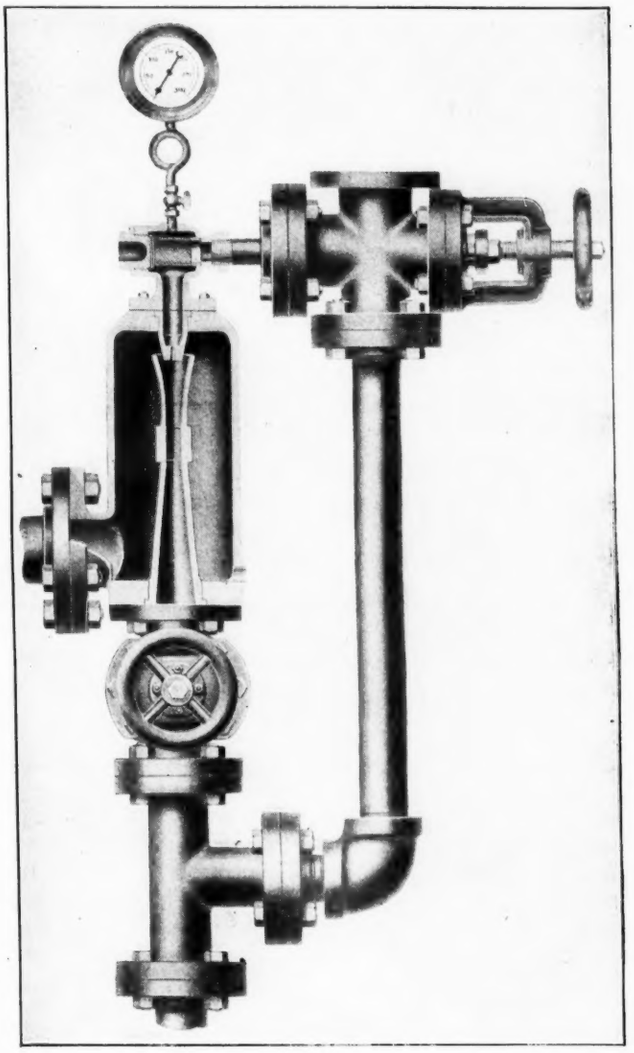
A glance at the differential draft gage showed whether the fans should be speeded or throttled. The fan regulation was by hand. When the differential gage read higher than the stoker indicator, the fires were in bad shape and capacity was being maintained at the expense of good combustion and efficiency.

G-R Regenerative Compressor

This regenerative compressor is designed for installation on Reilly evaporators, and its use during periods of reduced evaporator capacity permits of the evaporator operating at an efficiency much greater than can be secured by the evaporator operating alone, single effect. This is accomplished by compressing and delivering to the evaporator coils a portion of the vapor taken from the evaporator which, together with a certain amount of boiler steam, makes up the total steam required for evaporator operation. Without the use of the regenerative compressor all this steam would have to be supplied by the boiler.

The regenerative compressor, which is manufactured by the Griscom-Russell Co., 90 West St., New York City, is essentially a jet flow compressor that entrains and compresses, by means of live steam, a part of the vapor produced by the evaporator, and this entrained vapor is in turn led to the evaporator coils together with the boiler steam.

High-pressure steam from the boiler (not to exceed 250 lb. per sq.in.) enters at the steam inlet, the bypass valve being closed, but allowing the steam to enter the compressor through the strainer and then downward



SECTION THROUGH REGENERATIVE COMPRESSOR

through the steam nozzle. The steam leaves this steam nozzle at a high velocity and then enters the diffuser nozzle.

Part of the vapor from the evaporator passes through the side vapor inlet into the body of the compressor and then through the annular space between the steam nozzle and the diffuser nozzle.

By reason of the high velocity of the steam passing from its nozzle, the vapor is drawn into the diffuser nozzle and compressed to a pressure considerably above that of the vapor in the evaporator. The mixture of steam and vapor passes from the diffuser nozzle through the valve and fitting at the bottom and into the coils of the evaporator, where it gives up its latent heat to the brine in the evaporator shell, thus producing vapor and completing the operation.

Motor Drives For Air Compressors

Types of Air Compressor, Their Limitations, Power Requirements and Operating Characteristics—The Motors To Use for Different Applications

BY GORDON FOX*

FOR purposes requiring large volumes of air at low pressures up to 12 oz. per sq.in., centrifugal fans principally are used. To furnish large volumes of air at pressures ranging from 1 to 12 lb. per sq.in., positive-pressure blowers are well suited. They are seldom used for pressures above 12 lb., as the leakage within the blower becomes prohibitive. For the higher pressures turbine blowers or reciprocating compressors are used. Turbine blowers are inherently high-speed machines. They are best suited to moderate pressures because they develop about 3 lb. per stage and require a sufficient number of stages to make up the total compression. Turbine blowers, driven by steam turbines, are in common use in the large sizes, particularly in the steel industry. Centrifugal compressors of the smaller sizes are frequently motor-driven. Either a squirrel-cage induction motor or a direct-current shunt motor is used, and in either case the equipment is of the high-constant-speed direct-connected type.

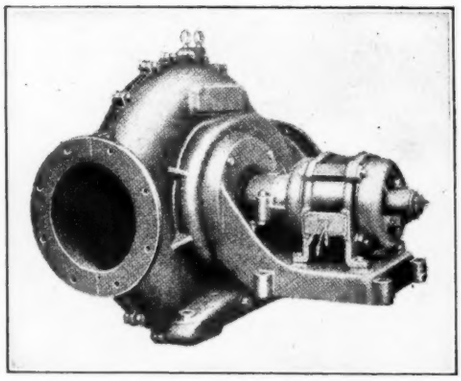


FIG. 1. CENTRIFUGAL COMPRESSOR DRIVEN BY INDUCTION MOTOR

Centrifugal compressors are essentially constant-pressure outfits, whereas positive-pressure blowers are constant-volume machines. Centrifugal compressors are adapted to service requiring a wide range of air delivery at fairly constant pressures of low values. Their advantages, as compared with reciprocating compressors, are high efficiency, simplicity, small floor space and low maintenance cost. The power requirements of a line of single-stage motor-driven centrifugal air compressors are shown in Table I.

TABLE I. POWER REQUIRED BY CENTRIFUGAL AIR COMPRESSORS

Pressure Lb. per Sq.In.	Free Air, per Min., Cu.Ft.	Hp.
1	800	5
1	1,600	10
1	3,200	20
1	4,500	30
1	7,200	50
1	10,200	75
2	750	10
2	1,600	20
2	2,500	30
2	4,200	50
2	6,200	75
3½	1,250	30
3½	2,400	50
3½	3,800	75
3½	9,000	175
3½	18,000	350

The great majority of air compressors are of the reciprocating displacement type, which is suited particularly for moderate volumes at fairly high pressures. For pressures up to 50 or 60 lb. per sq.in., simple, single-cylinder compressors are used. For higher pressures two-cylinder compressors, called two-stage, are better adapted, and very high pressures call for three-stage machines. Of the multistage compressor the principal advantage lies in the fact that the air is cooled between stages, reducing the volume necessarily handled in the subsequent stage, lessening the heating of the air and the consequent shrinkage in delivered air which occurs upon cooling. Multistage compressors are somewhat more efficient in their range than single-cylinder compressors. Power requirements to compress air depend largely upon the extent the air is cooled while under compression. Isothermal compression, with the air temperature maintained constant, is more efficient than adiabatic compression in which all heat remains in the air. Actual practice lies between these two. The mechanical efficiency of reciprocating compressors is about 85 to 90 per cent. Heat and other losses bring down the over-all efficiency to about 60 to 75 per cent. The horsepower required varies with different designs and sizes, but the figures in Tables II and III closely apply. Some of the larger multistage compressors require a power input as low as 10 to 12 hp. per 100 cu.ft. of free air for 100-lb. pressure.

TABLE II. HORSEPOWER REQUIRED BY RECIPROCATING AIR COMPRESSORS

Air Pressure, Pounds	Hp. per 100 Cu.Ft. Free Air per Min.
10	5
15	6
20	7.5
25	8.5
30	9.5
50	13–14
75	15–17.5
100	17.5–20
125	19–23

TABLE III. POWER REQUIRED TO COMPRESS AIR

Discharge Pressure Lb., Abs.	Kilowatt-Hour per 1,000 Cu.Ft. of Free Air	
	Single-Stage	Two-Stage
45	2.1	
65	3.0	2.98
85	3.7	3.2
105	4.3	3.7
125	4.8	4.1
145	...	4.4
175	...	4.8

Reciprocating compressors operate at low speeds, ranging from 80 r.p.m. in large sizes to 250 r.p.m. in small sizes. They are operated usually at constant speed and driven by constant-speed motors. The smaller sizes are commonly belt-driven, an idler pulley being used to provide good belt contact, with close centers and a rather wide speed ratio. Fig. 2 shows a drive of this type. A chain drive is sometimes used for small units. The larger units employ direct-connected motors, which results in a more compact installation.

*With Freyn, Brassert & Co., Engineers, Chicago.

Small compressors may be driven either by direct-current motors or by induction motors. Compound winding is preferable in the case of the direct-current drive. The great majority of compressors are driven by alternating-current motors, owing to the constant-speed characteristics and the usual undesirability of converting the power to direct current. For medium-sized compressors the slip-ring induction motor is often the preferable type. Such a machine can start the compressor under load, with automatic control, if desired. In starting it requires less current inrush than a squirrel-cage motor, often an important feature, inasmuch as most compressors require comparatively large motors.

For small and moderate sizes, the induction motor drive by belt is a satisfactory installation and has in its favor low first cost. For larger compressors direct-connected synchronous motors are desirable. They have higher efficiency and a high or corrective power factor. Besides, they eliminate belt losses, require less floor space and thus save in building cost. The synchronous motor maintains constant speed and will carry overloads without falling out of step. These considerations justify their higher first cost.

When driven at constant speed, reciprocating compressors are essentially constant-volume machines. Inasmuch as motor-driven compressors operate ordinarily at constant speed, some means is necessary to adjust the output to the demand. Where the demand is highly intermittent, this may be best accomplished by arranging the motor to start and stop, as required, to maintain the pressure within limits in a storage tank. Automatic magnetic control in connection with a pressure gage, governs the motor operation.

Where the demand is irregular but not intermittent, it is better to "unload" the compressor during periods of light demand. This consists in shutting off the valves so that no air is admitted, the cylinders being emptied through relief valves. A compressor operating unloaded requires about 15 per cent of its full-load input. Also, light-load operation over long periods is undesirable because of the low power factor in the case of an induction-motor drive. Here the synchronous motor has an added advantage. Air compressors are now available that do not unload entirely and suddenly, but by steps. These machines can be operated steadily at partial output, if desired, rather than intermittently, and they can be operated unloaded or with full output. This gives a more uniform motor load and lowers the power-demand peaks. Motors driving compressors equipped with unloading devices are limited as to loading and therefore require no overload capacity. In other words, drives of this type may be closely motored. Ordinarily, large compressors are arranged to start unloaded. The torque required for starting from rest is then about 30 per cent of full-load torque. This torque can be developed by properly designed synchronous motors. In so doing from 150 to 300 per cent of the full-load kilovolt-amperes is taken from the line. Self-starting synchronous motors for air-compressor drive should develop a pull-in torque not less than 50 per cent of full-load torque. A synchronous motor for this service requires a rotor winding of fairly high resistance to give sufficient starting torque, but should be able to pull into synchronism, with the field applied, while on the starting voltage. This minimizes the throw-over peak. A design giving maximum torque at about one-half speed meets the necessary requirements.

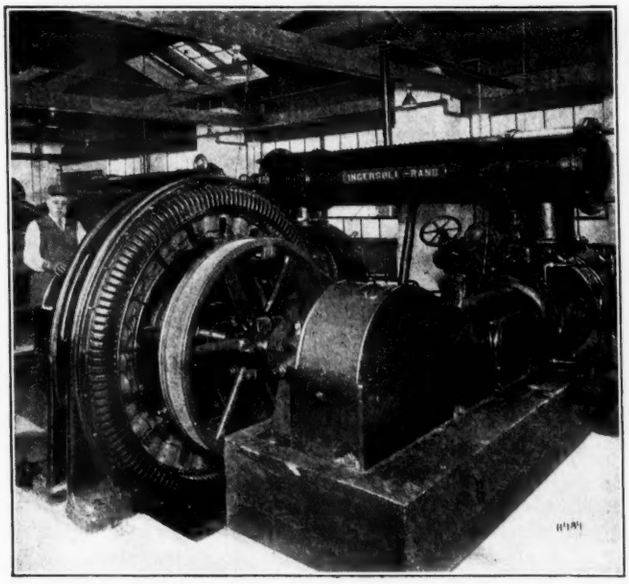


FIG. 3. TWO-STAGE RECIPROCATING COMPRESSOR DIRECTLY CONNECTED TO SYNCHRONOUS MOTOR

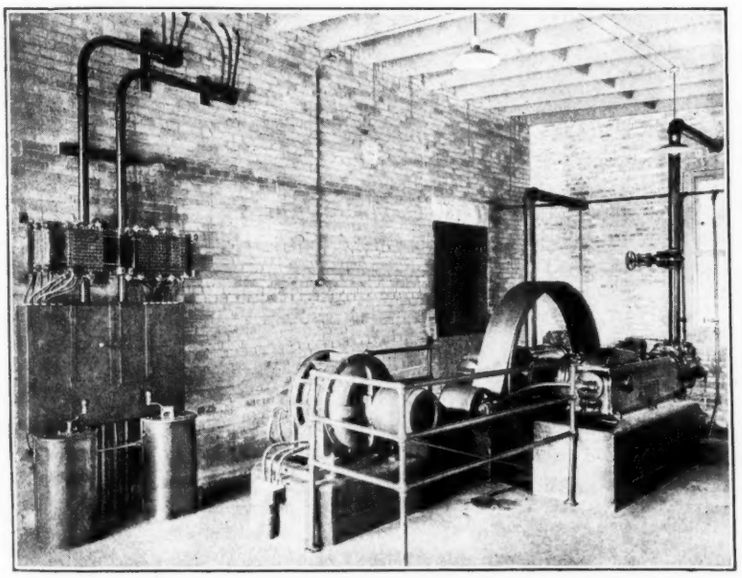


FIG. 2. SINGLE-STAGE RECIPROCATING COMPRESSOR BELT-DRIVEN BY MOTOR

For driving a reciprocating compressor, the torque required is irregular, having pulsations due to the varying crank effort. If the compressor is driven by a synchronous motor, this may tend to cause hunting. The amount of angular speed variation depends on the torque variation and the inertia of the revolving parts. The corrective torque is also of pulsating nature due to the variation of pole position in the motor. The variable motor torque resulting tends to oscillations, so that it is necessary to use a flywheel. The presence of a damping winding greatly reduces or eliminates entirely the tendency to hunting.

Centrifugal Pumps—Part II. Dependency of Efficiency upon Type, Size and Speed

By W. D. CANAN

IT IS the purpose of this article to show from actual tests how the efficiency of centrifugal pumps varies with type, size and speed. The data were obtained from reports covering a total of seventy-three tests published either in periodical or book form or in manufacturers' bulletins and in every case were read directly from the characteristic curves rather than the tabulated results. In the high-lift class the speeds range from 400 r.p.m. to 3,000 r.p.m., the capacities, 75 gal. per min. to 4,200 gal. per min. and the heads from 101 to 1,800 ft. The low-lift pump speeds range from 133 to 2,500 r.p.m., the capacities, 100 gal. per min. to 5,200 gal. per min. and the heads 17 to 100 ft.

In order to make a comparison of the various pumps, it was necessary that they be rated under similar conditions, and for this purpose the point of maximum efficiency was selected. This necessitated re-rating practically every pump, for it was found that the guaranteed, or manufacturer's rating in very few cases was the same as that corresponding to the point of maximum efficiency on the performance curves. In some cases the curves show more efficient operation and greater capacity, and in others greater efficiency with less capacity than the guaranteed conditions. In addition to the rating at the point of maximum efficiency, the operation at constant-speed condition was obtained for 25, 50, 75 and 125 per cent capacity or load.

Information relative to the effect of varying speed conditions could not be obtained in nearly the detail or quantity as that relative to varying capacity at constant speed, for the reason that about 95 per cent of tests are run under constant speed conditions. Three tests were found, however, that gave fairly good curves and the data tabulated in a manner similar to that previously described. The various speeds were calculated in percentage of that at which the pump operated at maximum efficiency and the head, capacity, brake horsepower and efficiency in per cent of rated values.

Method of Plotting Curves

Having collected all the data and listed them as described, efficiency was plotted against gallons per minute in order to find if there seemed to be a definite relation between the efficiency and the capacity. For this first plotting all pumps were taken regardless of type, head, etc., and the results showed that while the efficiency increased with the capacity, a great many of the points varied from 8 to 10 per cent on either side of the average curve. This fact tended to show that the efficiency was dependent to some extent upon some factor other than the capacity, although this seemed to be the principal one.

In an article on centrifugal pumps published in the *Engineer* six or eight years ago, is a statement to the effect that the efficiency of centrifugal pumps varies with the water horsepower. Using this idea, the water horsepower for all the pumps was calculated and plotted against the efficiencies. This plotting gave very poor results, there not seeming to be any relation between the two quantities. However, this may have been due to the fact that although the pumps selected covered a wide range of capacities, the water horsepower came within very narrow limits, about 90 per cent of them ranging up to 200 and the remainder scattered from this point to 800. No satisfactory curve could therefore be drawn.

Next it was decided to check the efficiencies as read from the performance curves with those calculated by dividing the water horsepower by the brake horsepower. Making allowances for mistakes in interpolating the values, the two efficiencies were substantially the same except in the case of one line of pumps for which in every instance but one the efficiency calculated from $\frac{W.\,Hp}{B.\,Hp}$ was from 12 to 14 per cent lower than that read from the performance test curves. Consultation of various authorities gave only one hint that might explain in part this discrepancy. Three methods are used in calculating the total head against which a pump works:

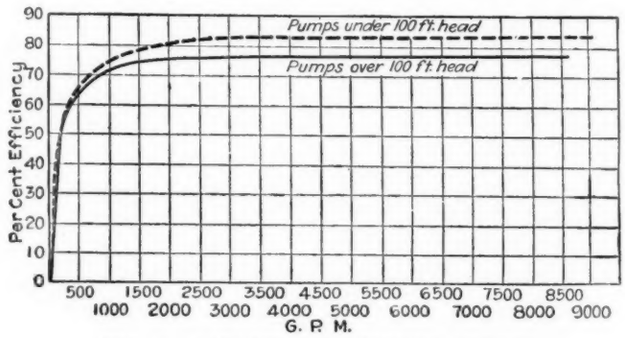


FIG. 1. CAPACITY-EFFICIENCY CURVES

First, by taking the sum of the discharge and suction heads and adding to them the vertical distance between the two gages; second, by adding to the head as found in the first case the velocity head $\frac{V^2}{2g}$ in the discharge line; third, by adding to the head as found in the first case the difference between the velocity heads in the discharge $\frac{V^2}{2g}$ and suction $\frac{V^2}{2g}$ lines. Now if the "total lift head" as plotted in the performance curves is calculated by one of the methods cited and the head used for calculating the water horsepower by another, a procedure that has been found to be employed in some tests, it would explain in part this difference. However, under the worst conditions this would not account for more than half of the discrepancy. The foregoing fact, together with superior workmanship and an error in interpolation, is the only explanation that can be given for this difference between the calculated efficiency and that read from the curves.

The third plotting of efficiency against capacity was made on the basis of high- and low-lift pumps, or for heads up to and including 100 ft. and for heads over 100 ft. This made it possible to separate the two classes

*The first part of Mr. Canan's article, treating of "Elementary Theory and Classification," appeared in the May 3 issue.

of pumps and gave opportunity to compare the efficiency of one class with the other. Fig. 1 shows the curves obtained from this last plotting.

It is just as important to know the efficiency of pumps at fractional and overloads as it is at full load. The efficiencies for 25, 50, 75 and 125 per cent loads taken from the test curves, were translated into terms of the 100

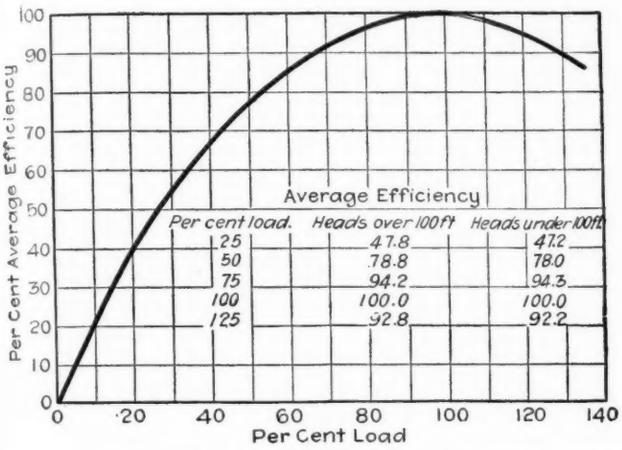


FIG. 2. AVERAGE EFFICIENCY-FRACTIONAL LOAD CURVE

per cent load or maximum efficiency by dividing the efficiency at the fractional or overload in per cent, by the maximum or full-load efficiency in per cent. Upon being tabulated, it was noticed that the values for the different pumps for the same load seemed to be rather uniform, so the points were plotted as shown in Fig. 2, percentages of load as abscissas and percentages of average efficiencies for the various loads as ordinates.

The last consideration is that of the speed and efficiency. As was said before, the majority of tests on centrifugal pumps are made with the speed held constant, owing to the fact that this is the condition under which the machine is most commonly used. Of the few available records of tests at varying speeds the conditions under which they were made were by no means uniform. In one case the head was held at constant value during the test, in another the capacity, and in the third neither head nor capacity was regulated in any way. Fig. 3 shows the results of three such tests as mentioned, the abscissas being percentages of normal speed, and the ordinates percentages of maximum efficiency. The curves show speed variations when the head is constant, when the capacity is constant and when both vary.

Capacity and Efficiency

The curves of Fig. 1 show, first, that the efficiency varies with the capacity, increasing very rapidly for the high-lift pumps from 75 gal. per min. to about 1,500, and for the low-lift from 100 to 2,000 gal. per min. Beyond this point the efficiency increases very gradually with the capacity following approximately a straight line. The greatest variation of any point from the curve is about 6 per cent, and considering the fact that there are represented pumps from fifteen different manufacturers whose standards of workmanship will vary greatly, this does not seem excessive. The efficiency of the low-lift pumps is greater than that of the high, the difference being noticeable for small capacities and becoming greatest at about 3,000 gal. per min. The actual values of the efficiencies for the two types of pumps as shown by the curves are as follows:

Gal. Per Min.	Over 100 Ft.	Under 100 Ft.
100	39.0	29.0
500	63.5	66.0
1,000	71.5	74.0
1,500	74.5	78.0
2,000	75.5	80.2
3,000	76.0	82.5
4,000	76.3	82.8

Fractional and Overload Efficiency

Fig. 2 shows how the efficiency varies for changes in capacity, the interesting feature of this curve being that efficiency for the fractional loads will be approximately the same regardless of the size or the type of the pump. The tabulated data represent the average of all tests from which the curves of Fig. 1 were obtained. It should be noted that the values for average efficiency are not actual efficiencies, but are percentages of maximum efficiency which were obtained under test conditions.

Efficiency—Speed

The curves shown in Fig. 3 differ from the two previous curve sheets in that in the former the curves represent the average of many points, while in the latter each curve represents values from one test only. Curve *A* shows the maximum efficiency variation with changes in speed, the head being held at a constant value. The curve rises and falls very rapidly, indicating that a small change in the speed will cause a great change in efficiency. For instance, a drop of 8 per cent in the speed will cause a corresponding drop of 10 per cent in the efficiency, while a drop of 10 per cent in the speed will lower the efficiency 25 per cent. Such a curve is to be expected with operation of this kind (that is, constant head with varying speed) because with a small change in speed there must be a relatively large change in the capacity in order to maintain the head. For instance, in the curve just referred to a drop of 8 per cent in

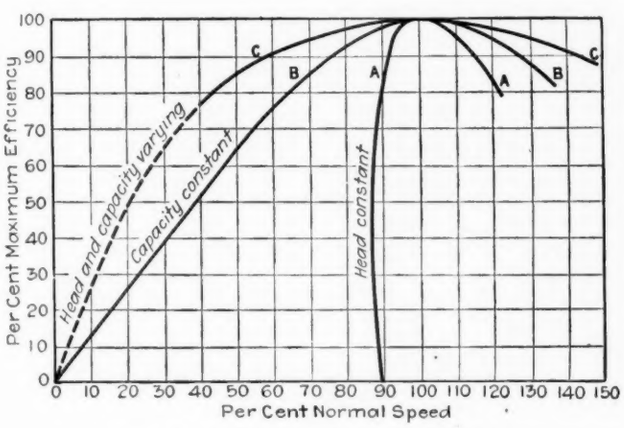


FIG. 3. EFFICIENCY-SPEED CURVES

the speed reduces the capacity 25 per cent and a drop of 11 per cent in the speed reduces the capacity 50 per cent.

It will be noted that this curve does not continue to the origin, but strikes the *X* axis at about 89 per cent speed. This means that with no output, consequently zero efficiency, 89 per cent of normal speed is necessary to maintain the rated head. Therefore no great variation of speed is possible if constant head must be held. The convex shape of the curve is due to

the design of the impeller. Following are some of the values of capacity and efficiency for various speeds:

Speed, Per Cent	Head, Capacity, Per Cent	Efficiency, Per Cent
87.3	25	32.7
89.0	50	69.3
92.5	75	92.2
111.0	125	94.3

All the foregoing values are in percentage of operation at the point of maximum efficiency.

Curve *B* shows changes in efficiency due to changes in speed, the capacity being held constant and the head allowed to vary. Here we have better operation than in the first case, it being necessary for the speed to change 40 per cent to reduce the efficiency 25 per cent. The speed may vary as much as 25 per cent above or below normal and still not cause a reduction of more than 10 per cent in actual efficiency. This increase in efficiency is due to the fact that more energy is used for lifting water and less for producing pressure head than in the previous case. The curve started at the origin because with zero head we have zero speed and efficiency. The values used in plotting this curve are as follows:

Speed, Per Cent	Head, Per Cent	Efficiency, Per Cent
25	2.2	32.0
50	14.3	63.4
75	44.9	88.8
125	....	88.2

Curve *C* shows the relation of the maximum efficiency and speed when both the capacity and head are allowed to change. This curve shows the least change in efficiency for a given change in speed of any of the three. For instance, an increase or decrease of 40 per cent in the speed will lower the efficiency only about 10 per cent. No points were available for speeds less than 80 per cent, but the curve was continued to zero, it being evident as in the case of curve *B*, that this will be the starting point. The reason that curve *C* shows the most efficient operation is that for every speed of a centrifugal pump there is one head and capacity at which the pump will operate at the maximum efficiency for that particular speed. The pump will adjust itself to these conditions if it is not prevented from doing so by some external means. However, if when the speed changes the pump is forced to deliver water at a different rate or head than it would inherently, then it will not develop the maximum efficiency for the particular speed as curves 1 and 2 show. The following table shows the efficiencies for the three cases for the same change in speed:

Speed, Per Cent	Curve 1, Per Cent	Curve 2, Per Cent	Curve 3, Per Cent
60	....	75.0	90.0
70	....	84.5	94.2
80	....	92.5	97.0
90	80.0	97.5	99.0
110	95.0	98.8	99.0
120	83.0	95.0	97.2

The following general conclusions may be drawn from the results herewith presented: First, the maximum efficiency of pumps working at heads under 100 ft. and over 500 gal. per min. capacity will in general be slightly higher than that of pumps working at heads over 100 ft. and of the same capacity; second, the percentage of this maximum efficiency which will be realized at fractional and overloads will be approximately the same for all pumps, regardless of type, size or head; third, when operating at other than normal speeds, attempting to maintain a constant head while permitting the capacity to vary, results in a large drop in efficiency for a small change in speed. The most efficient operation will result from allowing the head and capacity to adjust themselves to the changed speed conditions.

The Properties of High-Pressure Steam

By M. J. Eichhorn

Data available on the subject of the properties and behavior of steam at pressures ranging from 500 to 1,500 lb. per sq.in. are as yet comparatively meager, owing to the difficulties of experimentation. The steam tables of Marks and Davis extend to 600 lb. per square inch, except for the last part of Table I, which is brought up to the critical point, there given as 3,200 lb. per square inch. The "New Steam Tables" compiled by Smith and Warren (New York, 1912) give some of the properties of steam up to a pressure of 300 lb. per square inch. These tables are based on the formulas of Prof. H. L. Callendar, and contain in the introduction a résumé of the thermodynamical derivation of these formulas. The latest tables available are those calculated by Professor Callendar himself and extend up to pressures of 500 lb. per square inch, except for the last part of Table III of Professor Callendar's work, which stops at 653 lb. per square inch. The range of pressures covered in "Peabody's Steam Tables" is below 300 lb. per square inch. In addition to these more extensive tables there is in "Hütte" a short table giving the results of Holborn, Henning and Baumann in regard to the relation between the pressure and temperature of saturated steam in metric units, up to the critical point, which is there given as 3,189.3 lb. absolute per sq.in.

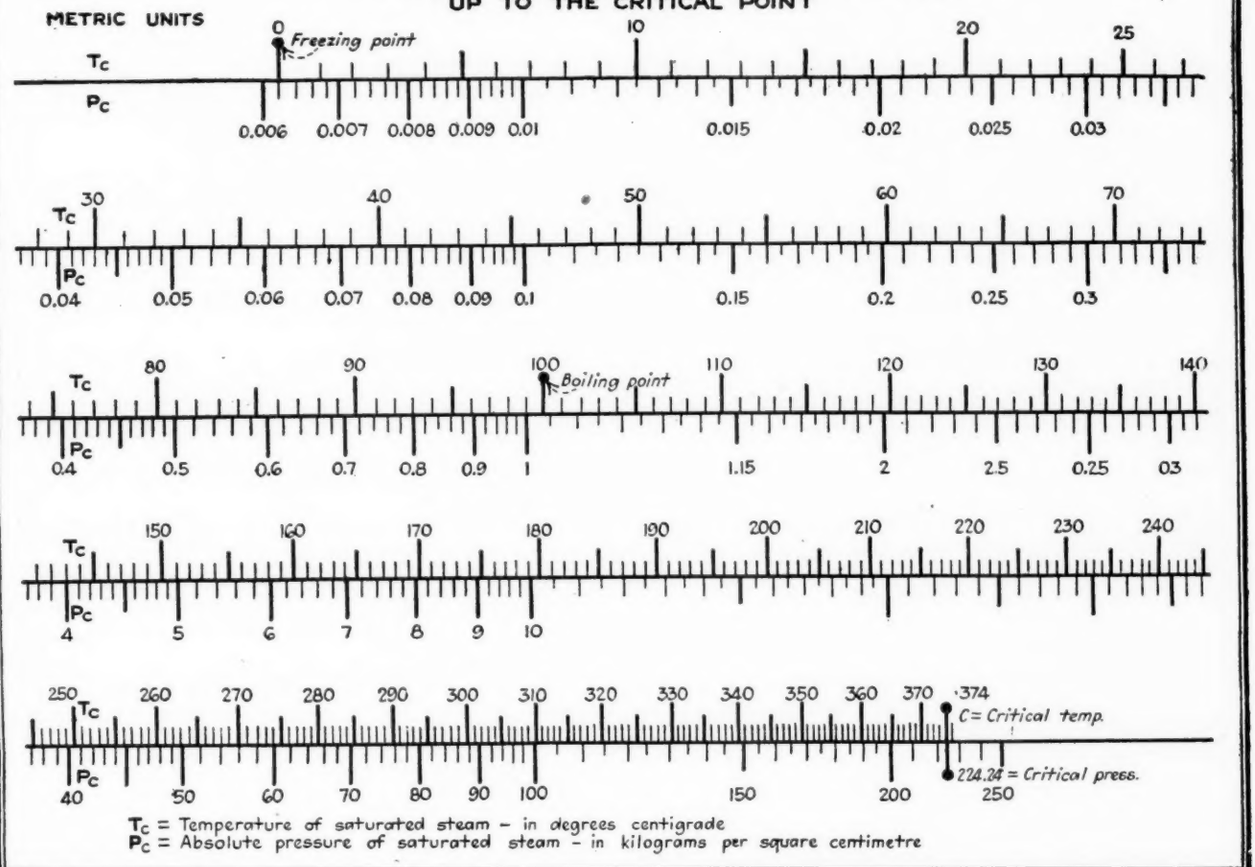
I have plotted this table in the form of a nomogram of two variables, for both metric and English units, which appears on the opposite page. In using the nomogram the only thing necessary to remember is that each point on one of the horizontal lines represents a pair of values; that is, one temperature and one pressure, which belong together and represent one particular state of the saturated steam. The five horizontal doubly divided scales in the nomogram are simply to be considered as a single nomogram divided into five parts, which are arranged one above the other. The reason for this arrangement is that the base scale is a logarithmic scale for the pressures, which in this case have approximately five decimal ranges; for instance, (1) 0.06 to 0.5, (2) 0.6 to 5, (3) 6 to 50, (4) 60 to 500, (5) 600 to 5,000 on the pressure scale.

The particular advantage I derived from this arrangement is that when once the upper group of scales has been plotted from the available table in metric units, then the base scale for the five lower lines (pressure scale) is the same as for the upper five, only displaced sideways to an amount corresponding to the constant of reduction. In this manner the point representing one pound per square inch on the lower group of scales occupies the same relative position as the point 0.0703 kilogram per square centimeter on the upper group of scales. In the same manner the temperature in degrees Fahrenheit for the lower group of scales is plotted from the centigrade scale above, thus making it unnecessary to calculate a numerical table for the English units, which would have involved a considerable amount of tedious arithmetical work.

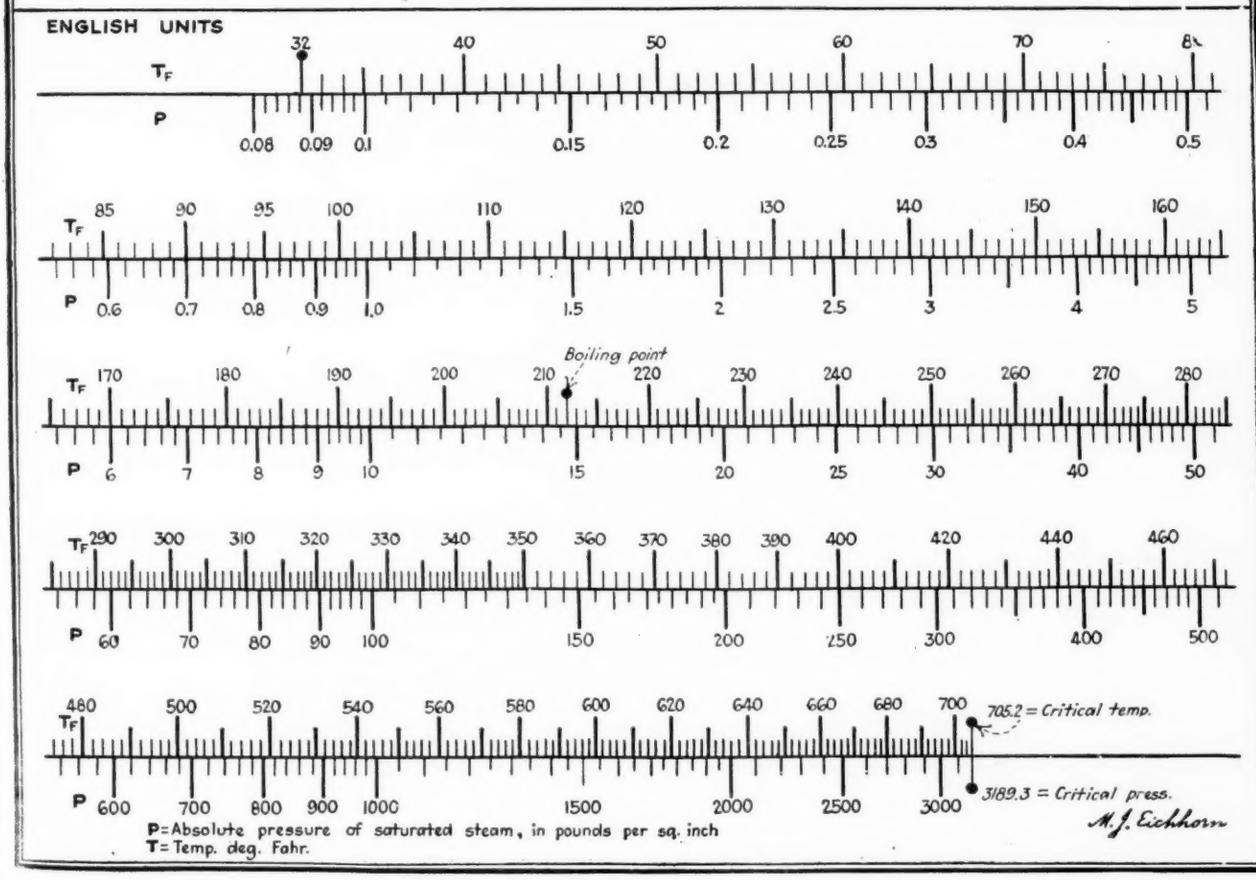
It is possible to obtain the English equivalents for the metric values of T_c and P_c for any given condition of the steam by dropping a vertical line downward to the English chart, reading the temperature and pressure directly from the proper scale.

Nomograms covering the total heat of saturated steam, entropy of saturated and superheated steam will appear in succeeding issues of *Power*.

NOMOGRAM FOR PROPERTIES OF STEAM
UP TO THE CRITICAL POINT

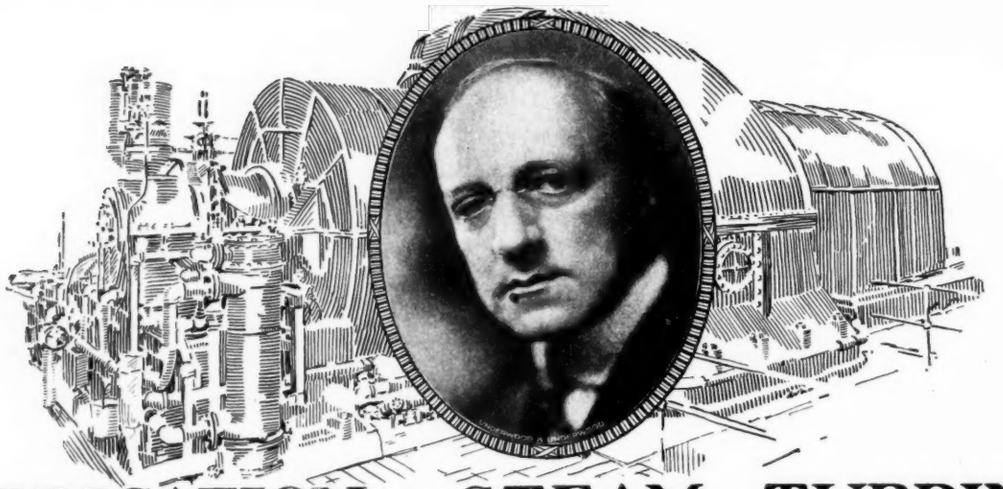


T_C = Temperature of saturated steam – in degrees centigrade
P_C = Absolute pressure of saturated steam – in kilograms per square centimetre



P=Absolute pressure of saturated steam, in pounds per sq. inch
T= Temp. deg. Fahr.

M. J. Eichhorn



LUBRICATION OF STEAM TURBINES

By WILLIAM F. PARISH*

AN ATTEMPT was made to lubricate one of the first steam turbines built by Sir Charles Parsons in England with a compounded steam-cylinder oil. When the turbine, which was one of the first commercial machines turned out, was ready for its shop trial, opinions were sought as to the kind of oil that would be best for the machine. It was thought that the temperature of the steam should be the determining point for the oil, and so this oil was selected and an attempt made to use it. The bearing temperatures were so high that it could not be used, and a heavy machine oil was substituted.

During the early 80's Tower, the English authority, and Thurston had been conducting experiments regarding the most efficient methods of lubrication, and their findings were widely read and discussed. Both of these investigators had determined from their experiments that force-feed or flood lubrication produced the best mechanical results. The turbine is singularly adapted for force-feed lubrication. The earliest designers had little difficulty in preparing the machine for what they considered a perfectly scientific and adequate lubricating system. They simply cast a compartment in the base of the machine and put in feed and overflow pipes and a rotary pump. So the first turbines embodied an entirely new principle of power generation to operate under many unknown conditions, lubricated by a system that is today considered the most efficient in principle.

Suitable Oil a Missing Factor

The one factor that was missing in the early days was suitable oil. Mineral oils were being developed that were fast displacing the animal and vegetable oils. No machine had been designed up to that time that depended upon an inclosed circulating and force-feed system of lubrication. Difficulties were experienced with these early turbines due to the fact that the system and the machine developed a series of conditions that rapidly brought the oil into a state where it would not work. When these troubles first occurred, there had been but little experience with similar conditions to use as a precedent for making recommendations. The oil men were not of the same opinion as to the source of the trouble. Some thought that the fault lay in the use of an oil that was too light; others took the opposite stand and maintained that the oils were too heavy.

In looking back over the years of constant effort to have turbine manufacturers design their machines and lubricating systems so that the lubricating oil would be given a chance to function properly, it is interesting now to record that the main difficulty always encountered with poor operating turbines was water. Air in the systems produced a second set of conditions that broke down the oil. The combination of water and air, with the constant heating and cooling of a small supply of rapidly circulating oil that had not been especially manufactured to withstand these severe and altogether new conditions, caused the wrecking of many turbines.

Water a Trouble-Breeding Element

In a station in Germany with three new turbines, the statement was made that the oil taken out of one wrecked machine contained 30 per cent water. An examination was made at the State Laboratories at the request of the turbine manufacturers, and a claim for damages was entered against the company that supplied the oil in use. The engineers representing both the owners and the builders would not believe that it was possible for water to get into the circulating oil through the operation of the turbine. Did not the oil system operate under two atmospheres pressure? How, then, could water under one-quarter that pressure force its way into the bearings filled with high-pressure oil? How was it possible for steam to leak out of stuffing boxes that were properly made and packed, and leap through eighteen inches of air and get into the circulating oil that was working under pressure? The easiest thing was to come to the conclusion that the water had actually been shipped in the oil from the refinery, possibly to reduce its cost.

The particular case referred to was settled by taking samples of the boiler water and of water that was drained from the turbine and a sample of the water used at the refinery. These samples were analyzed at the State Laboratories. It was shown that the character of the boiler water was the same as the water drained from the bottom of the oil tank. The water from the refinery had many characteristics that were not present in the other two samples.

* President, American Society of Lubrication Engineers.

Lubrication engineers attached to the marketing organizations of the large oil-refining companies had a great deal to do with the gradual improvement of the lubricating systems and bearings of the turbines. The case of the first large horizontal turbines built by an American firm in England, directly after the original Parsons patents expired, shows the method of educating all concerned with the design and manufacture of these machines.

The turbine was first set up in the shop to be tested before the official shop acceptance trial. It could not be run more than a few minutes because the bearings heated so quickly. After each attempt at operation, the oil in the system would be drawn out and some other put in. Seven kinds of mineral oils were used, ranging from the heaviest steam-cylinder oil and the compounded cutting oils to the lightest transformer oil. Finally, a grand combination of all the oils in the shop was made up and used with the same unsatisfactory results. Failure to start the machine resulted in a call for the lubrication engineer of the company supplying the oil. The mixture was taken out of the system, which was then flushed out with thin transformer oil to remove all deposits and dirt.

New drain pipes from the bearings were then put in, having double the capacity of the original pipes. A plunger pump, operated by a 50-hp. motor, was installed on the supply side of the system in place of the small geared pump, and the tank was filled with dynamo oil of the best grade. Thermometers were placed on every bearing and a log sheet was made that called for readings of the thermometers every five minutes. A man was stationed at each bearing. When everything was ready, the turbine was started and put right through its preliminary runs. It was then given its acceptance trials as soon as the engineers representing the purchasers could be brought from the hotel where they had been waiting for days. When a bearing started to heat, the oil supply to that bearing would be increased several times by reducing the supply to the other bearings. This would bring down the temperature very quickly. All that was done on that job was to arrange for an adequate supply of oil of the proper body, put in an oversized pump and drain system, and at the first indication of heating, flood the bearing under high pressure. The oil used would be considered good practice today, being a neutral, high-class dynamo oil at 145 seconds viscosity (Saybolt) at 100 deg. Fahrenheit.

Early Troubles Due to Emulsification

It was an easy matter in the early days to get the wrong kind of oil for the lubrication of turbines. When one of the first installations of the vertical-type machine widely used in this country some years ago was ready to start, the engineer in charge sent to a local oil warehouse for several barrels of the best oil they had. The warehouseman's idea of the best was the most expensive, and he therefore sent a high-grade marine-engine oil, properly compounded so that it would immediately form an emulsion with a small amount of water. One of the objectionable features and one which had much to do with the vertical type of machine being abandoned, was the fact that the stuffing boxes could not be kept tight.

The water and the marine-engine oil immediately formed a mixture that looked like a good grade of milk. Fortunately, the machine was stopped in time to prevent any damage and a new supply of more suitable oil secured. These vertical turbines gave a great deal of trouble due to water leaking from the stuffing boxes and getting into the lubricating systems. The water, nearly at boiling temperature, would readily mix with the air-filled, heated oil, especially at the middle bearing, where steam would also leak in and tend to cook the mixture. This brought about the introduction of the thinnest of all the oils that have ever been used for turbine lubrication in the attempt to get a lubricating medium that would allow prompt settling out of the water. These oils greatly assisted the settling out of the water, but did not allow enough margin of safety in the operation of the turbine. Some of these vertical turbines in Paris gave quite a lot of trouble. The oil would emulsify and become so thick that it could not be handled.

The foot-step bearings were changed from oil lubrication to water in order to prevent damage that might be due to the thickening of the emulsified oil. It was found that all the trouble was caused by water, but from two different sources: First, the water leaking from the middle bearing, and second, water dripping from the steam pipes in the boiler room onto the full barrels of turbine oil. The water leaked through the heads of the barrels and dissolved the glue with which the barrels were sized. The addition of fish glue, even of good quality, did not materially assist the oil in resisting a close combination with hot water. Proper storage facilities for the new oil somewhat helped the situation, and putting in new middle boxes reduced the trouble, but never entirely corrected it.

A station in one of the Southern States, having two of these vertical turbines, operated satisfactorily for nearly a year. Then there was a gradual increase in the vibration of the shafts and a noticeable effect on the lubrication of the turbine, which was not at first charged to the loosened bearings. The emulsified lubricating oil had become too thick to operate in the system. It was removed and new oil put in, which lasted thirty days and then became so thick that it could not be used. Several oils were tried with the same results, and finally a lubrication engineer was called in. He put the plant back to its original form by having new bushings put in the middle bearing and instituting a draining program. The trouble had been caused by the admission of an excessive amount of water to the system.

Turbine Lubrication a Double Problem

Turbines present a double problem: They must be fitted with oil on the same basis as any other machine—thin oil for the tight bearings and thick oil for loosely fitting bearings with large clearances; but the oil in every case must separate quickly from water. All oils do not act alike in the presence of water. Some oils will for a time allow water to separate freely, but subsequent shipments of the same oil will not have this characteristic to the same degree. Some oils will work perfectly in a turbine for a long time, subsequent shipments of the same oil acting the same, and then trouble will be experienced without any apparent cause. The upsetting of an almost perfect condition will always be found to be due to the overbalancing of the proportions of water or air mixed with the oil. As a rule the thinner oils will allow water to separate quicker than the thicker oils. There is no question that in the past the water difficulty resulted in the use of oils that were too thin to do effective work. Yet with the water conditions present, giving the machine the oil its mechanical condition required, stoppage would occur due to the heavier oil

emulsifying, forming a thick lather, "livering up" and refusing to circulate through the lubricating system.

Here was an unbalanced condition—a machine so designed that the proper kind of oil could not be used to lubricate it, and the development of the refining of mineral oils not at a stage where oils were made that possessed the necessary body for all the mechanical requirements of the turbine and still have the essential characteristic of quickly separating from water without forming serious and dangerous emulsions and deposits.

A Costly Lesson at Sea

A few years ago the facts were published regarding the lubricating troubles experienced on a modern battleship equipped with turbines of about 40,000 hp. The oil in use was a light-bodied oil which was extensively used as a regular turbine oil. During the preliminary runs previous to the official acceptance trials, the turbine shaft bearings burned out and renewals had to be made while at sea. Before a survey of engineers was ordered, numerous bearings had to be rebabbitted, all without shore machine-shop equipment, and the extra and discouraging work of rebabbitting at sea quite demoralized the crew. A survey was ordered. Some of the engineer officers were of the opinion that the ship should be towed back to New York and a new lubrication system installed. One officer had been experimenting with a very heavy-bodied lubricant for the force-feed system of the reciprocating engines and had found that the oil had a remarkable way of allowing water to drain from it. He suggested its use as the cheapest thing that could be done under the circumstances. A shipment of the oil was ordered. The light turbine oil in the system was pumped overboard. The new, heavy-bodied oil was put in the system and the battleship came back to New York under her own steam and immediately afterward went on her speed trials and was accepted. The bearings that had been babbitted at sea with only the facilities of the ship's machine shop did not require further attention.

What happened was this. The first oil would have been satisfactory in a dry system. The water that got into this system was salt water from the cooling coils and there had been some rusting of the tanks, so that the rust formed a binder for the oil and the sea water. The emulsion of the salt water and light turbine oil stopped the flow of oil to the bearings. The bearings in consequence burned, and this continued to occur as long as that particular oil was in the system. The new, heavy oil did not emulsify with the salt water. It was so heavy in body that it filled up the increased clearances of the bearings. If this particular oil had not been developed and used in an experimental way in the fleet at just that time, in all probability one of our largest battleships would have been towed to New York for inspection and overhauling.

Another Similar Instance

There was a somewhat similar case some months later on one of the destroyers fitted with high-pressure turbines and low-pressure reciprocating engines, all connected to the same lubricating-oil system. This system was filled with what was then the contract turbine oil for the Navy. The destroyer was taken out by the builders for the preliminary trials and trouble developed with the bearings. The ship was brought back to the yard, and the bearings that had been damaged were rebabbitted. A second trial was then attempted. On this trip a section of one of the mainshafts was damaged, the babbitt ran out and the casing brazed on the shaft. A new section of shaft was put in, and the very heavy oil that saved the battleship was put in the system after the regular Navy turbine oil was pumped overboard. With this oil the ship completed her trials, was accepted and passed to the Government.

Now the history shows an interesting check on the lubrication record of the ship. In fitting out for sea service, the trial charge of heavy oil was pumped overboard and the original light-bodied oil was put in the system, in accordance with regulations which state that only contract oil shall be used. The ship then started out on her maiden trip. The same series of bearing troubles developed and she had to put back for repairs. The contract oil was taken out and a light-bodied oil from a third refiner was put in the system. The ship again started for sea, but had to put back for repairs more extensive than at any other time. The record of the ship was then looked up, and the heavy oil that allowed the ship to make her guarantees while in the builder's hands was put in the system after the light oil was pumped out. The ship went into commission and operated for years without any further trouble.

The history of this ship showed that the same basic facts existed and the same conditions caused trouble with the battleship and many land installations. Salt water caused the oil to emulsify, stopping the supply pipes and causing the bearings to burn out. The mechanical conditions of both the battleship and destroyer jobs required heavy oil. Formerly, heavy oils could not be produced that would work with water. The light-bodied oils that were available and standard for turbine work did not afford a sufficient margin of safety. The slightest interruption of the feed or of the film caused the bearing surfaces to come together; the local heating reduced the film to a point where it did not support lubrication and resulted in failure of the bearing and of the turbine.

Value of Research Work Demonstrated

This is a good illustration of a case where the research work of the oil refiner was completed just in time to be of the greatest value to a great mechanical development. As the contract oils for this service at that time were thoroughly inadequate, the two ships would have been indefinitely laid up, new engines installed, or a new system of lubrication designed, all of which would have cost thousands of dollars.

Turbines, then, require specific treatment as far as the fundamental character of the oil is concerned. It must be made so that it will not form an emulsion with water. This requires it to be the very best of the grades put out by the oil companies for this purpose. An oil that is not properly manufactured in every respect—that is, one that is made from inferior stocks or that acts badly in any of the processes through which it must pass before it is finally finished—should not be used as turbine oil. Every practical refiner knows how to select his oils for this service and how to maintain the high quality necessary. These oils are expensive to make, but for this service the very best is required. The proper oils may be grouped somewhat similarly to the motor oils and can be called light, medium, heavy and extra-heavy. Most of the turbines will be in mechanical condition requiring a medium oil, not so many will need the light oil, and the heavy and extra-heavy oils will take care of the turbines that have suffered some dam-

age, are out of alignment, operating under severe conditions, badly worn, or in an extraordinary condition.

In any case the utmost care must be exercised in keeping water out of the system. Turbines should be fitted with partial filtering systems, which are now made very efficient. If such a system is not installed, a regular program should be instituted to drain the lower part of the tank at stated intervals.

The development of all machinery passes through about the same cycle as to lubrication—from the most primitive state through the various stages of improvement to the most efficient method of lubrication. When machinery reaches a state of perfection, it will give long years of service and require little care. The turbine started with the most perfect system for using oil, the force feed. The development has been in balancing the oil to the system and to the mechanical conditions of the machine, and the systems have grown from the little cast-in compartment, almost an afterthought, through the cooler stage, to the place where the modern installation has its separate oil tanks, filters and pumps and sometimes coolers, heaters and separators. The mechanical development has been for the purpose of giving the oil a fair chance, and the oil development has been toward perfecting an oil that will make good.

Insulating the Suction Pipe of the Ammonia Compressor

By M. A. Kolb

In the refrigeration plant of the ammonia-compression type there is seldom any doubt as to the advisability of having the return pipe from the expansion coils to the compressor insulated with good covering when this pipe is short and indoors. But with some plants the return pipe is a long one and is outdoors. This occurs where

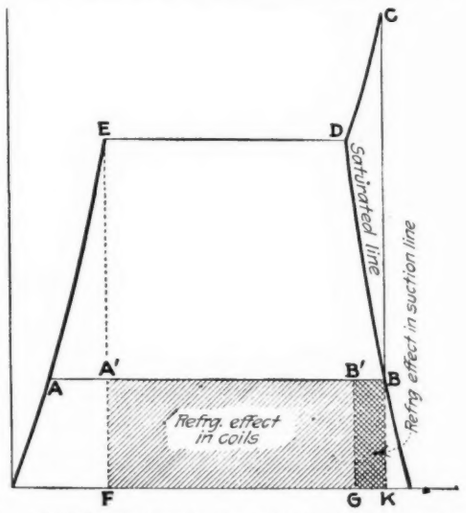


FIG. 1. REFRIGERATING DIAGRAM WITH SATURATED SUCTION VAPOR

one central plant supplies liquid ammonia to expansion coils in several distant buildings. In such cases the question as to whether or not the return line should be covered cannot be decided by any simple rules. It deserves and requires investigation.

The general conditions of operation with an insulated pipe would be quite different from those where a bare pipe is used. With a covered line sufficient liquid ammonia would be fed through the expansion valve to get the vapor back to the compressor in a cold saturated state. The temperature of the vapor and liquid in the return pipe would then be about the same as in the expansion coils, say about zero F. The temperature-entropy diagram for the refrigeration system with saturated suction would appear similar to Fig. 1. Here *AB* is the expansion or cooling-coil line with the ammonia

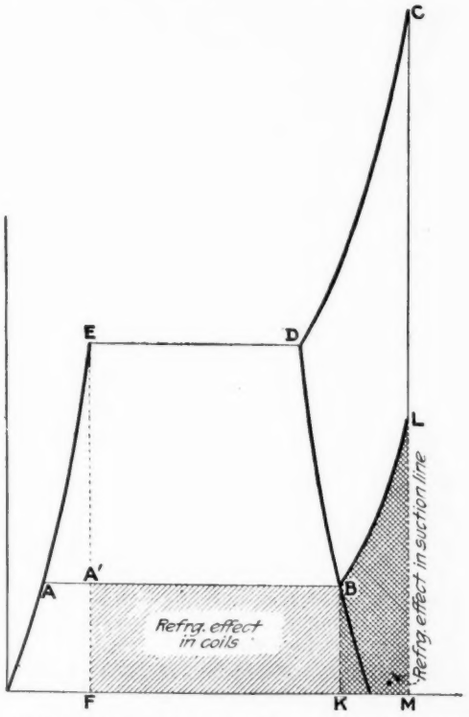


FIG. 2. DIAGRAM WITH COMPLETE VAPORIZATION IN THE COILS AND SUPERHEATED SUCTION

reaching the compressor in a saturated condition represented by the point *B*. The compression would be in a superheated state along the line *BC*. The cooling in the condenser would follow the lines *CDE*. The effective heat removal in the coils would be the shaded area A_1FGB_1, which extends to the 0 absolute temperature line, while the hatched area B_1GKB extending to the zero line represents the heat absorbed in the suction line. The area EA_1BCDE is the work done by the compressor.

With a bare pipe, however, the refrigeration loss would be too great to try to keep the vapor so cold, especially where a hot sun is shining on the pipe. So probably the best method of operation would be to feed just enough liquid ammonia into the expansion coils to keep only the coils cold and not the return line. In its travel back to the compressor the vapor would then soon become heated up to atmospheric temperature, or even a little higher if the pipe is a long one. The temperature-entropy diagram for this method of operation would be quite like Fig. 2. Here the ammonia expands to a saturated condition in the coils, the heat absorbed will be the area A_1FKB. The heat absorbed in the suction pipe would superheat the ammonia to *L*, about atmospheric temperature. The area *BKML* is the heat taken up, while the area $A_1BLCDEA_1$ represents the compressor work.

The two main objections to operating with a covered return pipe are: (1) The initial cost of the covering;

(2) the refrigeration loss due to operating with a cold suction pipe. It would cost several hundred dollars to properly insulate a pipe which is 400 or 500 ft. long. And in spite of this covering there would be an appreciable refrigeration loss, since the temperature difference between the atmosphere and the ammonia might easily be 80 deg. or 90 deg. F.

The chief objection to operating with a bare pipe is that the vapor at the suction of the compressor is superheated. Superheat rarefies the ammonia vapor and reduces the capacity of the compressor; and, owing to the high temperatures that it causes in the compressor, lubrication difficulties may be experienced. A superheated suction vapor also leads to a high temperature after compression, which throws an additional load on the condensers.

While all the points to be considered cannot be definitely evaluated, still some figures can be arrived at which may serve as a basis from which a decision may be reached. For instance, a yearly interest and depreciation charge for the pipe covering can be worked out from the initial cost of material and labor. The loss in refrigeration in operating with a cold, covered suction pipe can be reduced to a yearly charge by assuming or observing the average ammonia temperature in the suction pipe and the atmospheric temperature outside, by using a coefficient of heat transmission for the insulation taken from a handbook or a manufacturer's catalog and by using a predetermined figure for the cost of power per ton of refrigeration.

The loss per year due to a reduced compressor capacity caused by superheat can be taken as either a loss of power in running the machine or as a loss in production of the material being cooled. Just how much loss is incurred by using superheated suction vapor instead of saturated vapor can be ascertained by comparing the densities of superheated and saturated vapors, since the ability of a compressor to deliver pounds of compressed vapor depends directly upon the density of the vapor in the suction pipe of the machine. Tables on the properties of ammonia vapor can be used to find the densities at various pressures and temperatures. Thus, if we are operating with a suction pressure of 15 lb. gage and a temperature of 100 deg. F., the tables show the volume of the vapor as 11.55 cu.ft. per lb. Now, if the vapor were saturated and at the same pressure, the volume would only be 9.17 cu.ft. per lb. Hence, with the same power consumption the capacity of the compressor can be increased by 26 per cent by using saturated vapor instead of superheated vapor at 100 deg. F.

$$\left(\frac{11.55 - 9.17}{9.17}\right) \times 100 = 26 \textit{ per cent}$$

Or the amount of power that is being wasted by using superheated instead of saturated vapor is theoretically 21 per cent.

$$\left(\frac{11.55 - 9.17}{11.55}\right) \times 100 = 21 \textit{ per cent}$$

By taking these various factors into consideration, together with any outstanding data peculiar to a given installation, some light can be thrown on the question of covering the long outdoor return ammonia line. In attempting to operate the compressor with a saturated suction, wet compression should not be used, as it has been proved that wet compression is not as economical as when the ammonia vapor reaches the compressor in a dry condition.

Gas-Engine Crankshaft Alignment

By William P. Flint

A considerable number of vertical gas engines are in use that are provided with flywheels at each end of the crankshaft, as in Fig. 1. Such engines, if having three cylinders, usually are provided with two short bearings inside the frame and two long bearings which are placed at the ends of the frames. The crankshaft of this type of engine with three cylinders is shown in Fig. 2.

With such a design an important question of operating procedure arises. There is a question as to whether all the shaft bearings should be in perfect alignment.

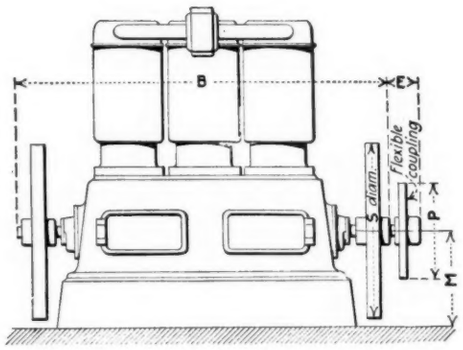


FIG. 1. ENGINE WITH OVERHANGING FLYWHEELS

If so, how is the engineer to know when this state has been attained on an engine without realigning the shaft? The usual method of adjustment is to slack off on the center bearings in the event that they have a wedge adjustment, level up the shaft at the two end bearings, and then bring the lower halves of the inner bearings against the shaft. When the bearings are in good condition, fair results can be secured by this method. Contrary, however, to the prevailing belief, the center line of the shaft is not straight when the bearings are so aligned. Owing to the leverage of the two flywheels bowing the shaft a small amount, the inner bearings when raised against the shaft are actually higher than the outer ones.

At one time the writer made some careful experiments with a 14 x 18 three-cylinder vertical gas engine quite like the design of Fig. 1. It was thought that with all four of the bearings in absolute alignment ideal running results would be secured. A very accurate straight-edge was set up parallel to the shaft center line, and a special leveling rod was made, having a spirit level and fitted with a micrometer adjusting screw at one end. This end rested on the straight-edge, while the other end was bent to allow it to enter the oil hole of the bearings and rest on the shaft, as in Fig. 3. After aligning the two outside bearings, it was a simple matter to measure the amount of bow in the shaft at the inner bearings. The maximum bow was 0.05 in.

The four bearings were next brought into alignment by lowering the center bearings and drawing down their top caps until the leveling rod read true. Contrary to expectations the adjustments did not give satisfactory results, for the inner bearings heated. When adjusted by being slacked off and then being

brought up against the shaft, the bearings had no signs of heating.

These experiments indicated the following conclusions:

First: The center bearings of a multi-cylinder engine having overhung flywheels should be higher than the outer bearings so that they may not exert a downward pressure on the shaft unless the shaft is sufficiently large to prevent deflection.

Second: Caution is required in giving clearance to the main bearing caps, since if insufficient clearance is allowed a downward pressure will be exerted at the inner edge of the cap on the shaft.

Third: New bearings require frequent adjustment while the shaft is wearing each bearing to what might be called its prevalent inclination.

Another question naturally arises. How is the engineer to know that he had made the adjustment he seeks? The first source of uncertainty is the condition of the main bearings. Experience shows that wear of the upper half of the main bearings should be negligible under normal conditions. When, however, they are allowed to heat, because of lack of oil or because of being keyed up too tight, the case is quite different. and the top shell may wear badly.

Frequently, an operator detects a hot bearing before the temperature seems excessive. He checks it off a little, floods it with oil and cools it down. Afterward he keys it up again as much as he dares and assumes that he caught it before the babbitt began to run. How is he to make sure without removing the bearing, and particularly how is his superior, who may not have heard of the trouble, to detect it?

Usually, when a bearing heats slightly, it is the babbitt in the lower box which softens and is carried up and deposited on the surface of the babbitt of the upper box. This layer may easily be a sixteenth of an inch or more thick, and makes it impossible to raise the shaft to its original height. The proof is very simple.

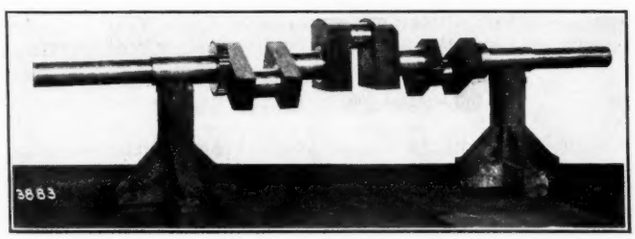


FIG. 2. THREE-THROW GAS-ENGINE CRANKSHAFT

If no such openings exist, holes should be drilled through the upper beam housing or cap and the distance from the top of the cap to the shaft measured and recorded. By measuring these distances each time the crankcase is opened, any lowering of the shaft can be detected. Too much clearance, rather than too little, is the usual danger signal.

In the foregoing no reference has been made to any belt pull. This may be in any one of several directions, and its magnitude may easily be considerable. The aim should be to provide for it in the same manner as for the flywheel weights. Whenever the belt pull has a horizontal component, the setscrews provided for holding the center bearings in lateral alignment, should be used to allow the shaft to take its natural curvature in the horizontal plane and to lock the center bearings in place.

In direct-connected units, with the engine shaft extended through a generator to an outboard bearing, the aim is usually to put comparatively little load on the outboard bearing, by locating it at the natural height of the end of the extended shaft, when the bearings of the engine proper are correctly aligned. These outboard bearings are usually small and well lubricated and are subject to but little wear. They should, however, be slacked off when the engine bearings are being keyed up and should afterward be adjusted to the natural height of the shaft.

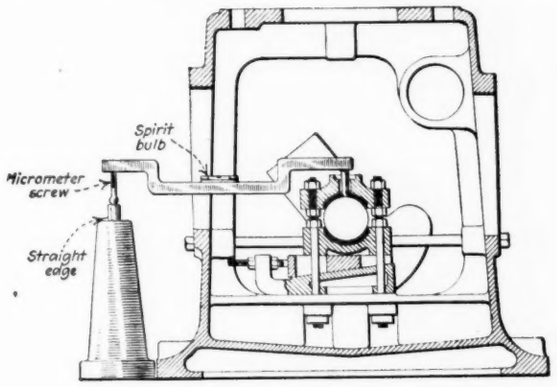


FIG. 3. CHECKING SHAFT ALIGNMENT

An Analysis of Power-Plant Costs

BY W. R. METZ

All power-plant engineers and employees know that costs have been mounting by leaps and bounds, but it is a question if the majority know the principal offender. It is for this reason that this article is written and an attempt made to show where the fault lies, in at least one plant.

The tabulated statement shows the itemized cost of operating and maintaining a plant under my supervision, having a total of 30,000 sq.ft. in boilers and an electric generating plant of about 2,500 kw. total capacity, about one-half of which is used. This plant is in operation 24 hours a day, 365 days in the year, and for this reason the machinery is in duplicate as far as possible.

In this statement the boiler-room labor covers firemen, coal passers and laborers; engine-room labor covers engineers, oilers, etc., and dynamo-room labor covers the switchboard operators and dynamo tenders. The labor for storing coal started in 1916, as previous to that time the contract for coal included storing, whereas after that time the coal was dumped only and had to be stored by our own force. The ash handling covers the amounts paid to contractors for hauling ashes away from the building.

Examining the tabulated statement in detail, we find as follows:

Boiler-room labor has increased from 0.173c. to 0.279c. per kw.-hr. or about 61 per cent. During this period helpers wages were increased from 35c. to 60c. per hour, firemen from 40c. to 50c. per hour and laborers and coal passers from 25c. to 35c. per hour. The average increase was about 40 per cent, so that the boiler-room

labor cost increase was largely due to increase in wages. The remainder may easily be credited to the increased cost of handling the fuel and the largely increased amount of ashes due to the poor grades of coal received.

Engine-room labor increased from 0.201c. to 0.236c. per kw.-hr., or about 17 per cent. In this part of the power plant, engineers' wages advanced from 50c. to 70c.

OPERATING EXPENSES OF POWER PLANT

Operation:	Average Cost per Kw.-Hr. 1915–16 Cents	1916–17 Cents	1917–18 Cents	1918–19 Cents	1919–20 Cents
Boiler-room labor	0.173	0.165	0.194	0.304	0.279
Engine-room labor	0.201	0.192	0.199	0.259	0.236
Dynamo-room labor	0.161	0.168	0.139	0.170	0.191
Fuel	0.446	0.470	0.749	0.854	0.733
Labor storing coal	.....	.....	.....	0.011	0.009
Engine oil	0.010	0.016	0.009	0.019	0.011
Cylinder oil	0.012	0.015	0.009	0.035	0.029
Waste	0.0002	0.0001	0.0002	0.0005	0.001
Boiler-room supplies	0.003	0.001	0.003	0.001	0.011
Engine-room supplies	0.015	0.004	0.006	0.009	0.015
Dynamo-room supplies	0.00	0.0002	.....	.....	0.0001
Office force and supplies	0.029	0.025	0.022	0.034	0.038
Ash handling	0.005	0.005	0.006	0.023	0.017
Total operation	1.0552	1.0613	1.3362	1.7195	1.5701
Maintenance and repairs:					
Building and fixtures	0.005	0.004	0.002	0.005	0.005
Boilers	0.047	0.041	0.076	0.064	0.054
Economizers	0.042	0.0002	.....	.....	0.001
Pumps	0.009	0.005	0.002	0.001	0.004
Auxiliary boiler room	0.006	0.005	0.006	0.015	0.008
Piping	0.023	0.012	0.013	0.021	0.009
Engines	0.032	0.035	0.014	0.012	0.046
Generators	.....	.....	.....	0.005	0.005
Condensers	.....	.....	.....	.....	0.006
Switchboard and meter	.....	.....	.....	0.0002	0.0002
Auxiliary engine room	0.001	0.004	0.001	0.008	0.001
Total maintenance	0.165	0.1062	0.114	0.1312	0.1392
Grand total	1.2202	1.1675	1.4502	1.8507	1.7093

Total kw.-hr.: 1915–16, 3,710,948.5; 1916–17, 3,782,718.2; 1917–18, 4,600,001; 1918–19, 4,558,559.4; 1919–20, 4,453,438.

per hour, oilers from 30c. to 45c. per hour and helpers from 35c. to 60c. per hour, the total average increase being about 52 per cent.

Dynamo room labor increased from 0.161c. to 0.191c. per kw.-hr. or about 18 per cent, whereas wages in this department increased over 50 per cent.

It can readily be seen from the foregoing that the boiler-room labor was the only item that increased disproportionally. Investigations showed that, owing to poor grades of coal furnished, the overtime of firemen increased enormously, and the labor for handling coal and ashes also increased. The items that were not affected by coal showed no increase beyond that occasioned by increase in salaries, but rather showed a substantial decrease. Summing all this up, it would appear that the main reason for increased cost in the power plant is directly chargeable against the poor quality of coal that was furnished.

During the same period the cost of the coal increased from 0.446c. to 0.733c. per kw.-hr., or about 64 per cent, which is readily explained by the present prices as compared to those in 1915 and 1916.

The total cost of operation will be seen to have increased from 1.055c. to 1.57c. per kw.-hr. or about 48 per cent. The cost of maintenance and repair decreased the first year, but after that showed a gradual increase due mainly to increased cost of labor and materials, but this increase is not at all out of proportion.

Were it not for economies practiced here and there, the total cost would have increased far beyond the actual increase, and it is to the credit of the engineering force that these costs have been kept down. But with coal evaporating around six pounds of water where good fuel formerly evaporated around nine or ten pounds, there can be no economy practiced.

Factor of Evaporation—Approximate Formula for Use with Saturated and Superheated Steam

By William F. Ryan

In an increasing number of boiler plants the engineer is required to record equivalent evaporation per pound of coal and other items that involve the so-called "factor of evaporation." While the calculation is a simple one, the use of steam tables and charts is inconvenient in the average plant.

The factor is frequently calculated to three and four decimal places, but the feed-water meters, recording pressure gages and recording thermometers from which the data for the calculation are obtained, are seldom sufficiently accurate to warrant such refinement. Two decimal places will ordinarily be sufficient.

With superheated or dry saturated steam the total heat varies little with changes in pressure, provided the steam temperature remains the same. The factor of evaporation, therefore, may be calculated with reasonable accuracy from steam temperature and feed temperature only. For pressures and temperatures usually found—that is, from 100 to 250 lb. gage pressure, 0 to 200 deg. superheat and 100 to 300 deg. feed temperature—the factor of evaporation may be found with sufficient accuracy from the following easily remembered formula:

$$F = 1.08 + \frac{T}{2000} - \frac{t}{1000}$$

Where F is the factor of evaporation, T the temperature of steam, deg. F., and t the temperature of feed water, deg. F.

The accompanying table evaluates the foregoing formula for the range of temperatures commonly found in practice, and is correct within 1 per cent. Calculations are based on Marks and Davis Steam Tables.

FACTORS OF EVAPORATION FOR SATURATED AND SUPERHEATED STEAM, BETWEEN 100 AND 250 LB. GAGE PRESSURE AND 0 DEG. TO 200 DEG. SUPERHEAT STEAM TEMPERATURE, DEG. F.

Feed Temperature, Deg. F.	340	360	380	400	420	440	460	480	500	520	540	560	580	600	620
100	1.15	1.16	1.17	1.18	1.19	1.20	1.21	1.22	1.23	1.24	1.25	1.26	1.27	1.28	1.29
110	1.14	1.15	1.16	1.17	1.18	1.19	1.20	1.21	1.22	1.23	1.24	1.25	1.26	1.27	1.28
120	1.13	1.14	1.15	1.16	1.17	1.18	1.19	1.20	1.21	1.27	1.23	1.24	1.25	1.26	1.27
130	1.12	1.13	1.14	1.15	1.16	1.17	1.18	1.19	1.20	1.21	1.22	1.23	1.24	1.25	1.26
140	1.11	1.12	1.13	1.14	1.15	1.16	1.17	1.18	1.19	1.20	1.21	1.22	1.23	1.24	1.25
150	1.10	1.11	1.12	1.13	1.14	1.15	1.16	1.17	1.18	1.19	1.20	1.21	1.22	1.23	1.24
160	1.09	1.10	1.11	1.12	1.13	1.14	1.15	1.16	1.17	1.18	1.19	1.20	1.21	1.22	1.23
170	1.08	1.09	1.10	1.11	1.12	1.13	1.14	1.15	1.16	1.17	1.18	1.19	1.20	1.21	1.22
180	1.07	1.08	1.09	1.10	1.11	1.12	1.13	1.14	1.15	1.16	1.17	1.18	1.19	1.20	1.21
190	1.06	1.07	1.08	1.09	1.10	1.11	1.12	1.13	1.14	1.15	1.16	1.17	1.18	1.19	1.20
200	1.05	1.06	1.07	1.08	1.09	1.10	1.11	1.12	1.13	1.14	1.15	1.16	1.17	1.18	1.19
210	1.04	1.05	1.06	1.07	1.08	1.09	1.10	1.11	1.12	1.13	1.14	1.15	1.16	1.17	1.18
220	1.03	1.04	1.05	1.06	1.07	1.08	1.09	1.10	1.11	1.12	1.13	1.14	1.15	1.16	1.17
230	1.02	1.03	1.04	1.05	1.06	1.07	1.08	1.09	1.10	1.11	1.12	1.13	1.14	1.15	1.16
240	1.01	1.02	1.03	1.04	1.05	1.06	1.07	1.08	1.09	1.10	1.11	1.12	1.13	1.14	1.15
250	1.00	1.01	1.02	1.03	1.04	1.05	1.06	1.07	1.08	1.09	1.10	1.11	1.12	1.13	1.14
260	0.99	1.00	1.01	1.02	1.03	1.04	1.05	1.06	1.07	1.08	1.09	1.10	1.11	1.12	1.13
270	0.98	0.99	1.00	1.01	1.02	1.03	1.04	1.05	1.06	1.07	1.08	1.09	1.10	1.11	1.12
280	0.97	0.98	0.99	1.00	1.01	1.02	1.03	1.04	1.05	1.06	1.07	1.03	1.09	1.10	1.11
290	0.96	0.97	0.98	0.99	1.00	1.01	1.02	1.03	1.04	1.05	1.06	1.07	1.08	1.09	1.10
300	0.95	0.96	0.97	0.98	0.99	1.00	1.01	1.02	1.03	1.04	1.05	1.06	1.07	1.08	1.09

EDITORIALS

Motor Drives for Air Compressors

IN A large majority of power applications the class of motor used is determined by the kind of current available; that is, either direct or alternating. Since air compressors usually operate at constant speed, either class of motor may be applied. Where direct current is available, the compound type of motor is preferable, since the series winding serves to iron out the peaks by giving the motor a drooping load-speed characteristic, and also its torque increases more rapidly than that of the shunt type, owing to the series-winding flux. Where the compressor is started and stopped frequently, the starting current will be considerably reduced with a given rate of acceleration by the use of compound-wound motors over the shunt type. Direct-current motors, either shunt or compound type, lend themselves more readily to automatic control, especially in the larger sizes. However, this feature is not of sufficient importance to warrant considering converting alternating current into direct, especially since alternator-current automatic control equipment has been developed to give a fair degree of satisfaction.

Between the use of either current there is on one hand the brushes and commutator of the direct-current motors, which always require considerable attention to keep the motor in good operating condition, and on the other there are the magnets of the alternating-current controller, which require no small amount of attention to keep them so that they will not only function properly, but also operate quietly. Where synchronous motors are used with automatic control, means must be provided not only for the control of the alternating-current circuit, but also for cutting in the exciter circuit at the proper time, which further complicates the control problem. However, where alternating current is available, synchronous motors, on account of their high efficiency and ability to operate at high or corrective power factors, are fast coming into wide application. Where frequent starting and stopping of the compressor is required and alternating current is available, the induction motor, either of the squirrel-cage or wound-rotor type, possesses a distinct advantage over the synchronous type on account of superior starting characteristics.

When starting is infrequent and the demand for air is taken care of by an unloading device on the compressor, the synchronous machine has a distinct advantage over the induction type. As pointed out by Gordon Fox in "Motor Drives for Air Compressors," in this issue, a compressor operating unloaded requires approximately fifteen per cent of its full-load power. An induction motor operating under full load, driving a compressor, will have a power factor of about eighty-seven per cent lagging, but when the compressor is unloaded the power factor will drop to approximately forty per cent, and even lower than this if the motor is operating at less than full load when the compressor is loaded. With the synchronous motor, if it is of the correct capacity to drive the compressor, the full-load power factor can be adjusted to unity. Even under this condition it may result in a material power-factor correction if the compressor load represents around twenty per cent or more of the load on the system. When the unloading device operates, the power factor of the motor automatically becomes leading and at fifteen per cent load will be approximately forty per cent. On most alternating-current systems this feature is a distinct advantage, since in general the power factor is lagging and some means of correcting is desirable. An underloaded induction motor on the system is always a liability, since it operates at a low power factor, and there is no way of correcting this without installing expensive auxiliary equipment or replacing the motor by one of the proper size. With an oversized synchronous motor the power factor can be adjusted to unity at the compressor load, and if desirable the excess capacity of the machine may be used to supply leading corrective component to the system, up to the kilovolt-ampere capacity of the motor, by overexciting the field coils. Therefore any excess capacity of a synchronous motor can usually be turned into an asset.

Need of a Rustless Steel

A STRONG non-corrosive steel is badly needed for many power-plant services. Corrosion has often proved so rapid in steel feed-water piping in large plants that it is now seldom used. Cast-iron or brass pipe is standard practice for such service. Cast iron must be made very heavy to stand the high pressures carried, and while less corrosive than steel, it seems an illogical material to use on account of its mechanical properties and the difficulty of taking up expansion in long lines. Brass pipe, while satisfactory in service, is expensive. It would seem that recent information on oxygen in feed water and methods of its removal should encourage metallurgists to develop a form of fairly rustless steel that could be used for boiler-feed lines in place of either cast iron or brass.

Increasing fuel prices have caused engineers to consider economizers to save some of the heat wasted in the flue gases. Cast-iron economizers are not favored for high pressures. Steel tube-economizers are subject to corrosion, largely internal but also external, unless means are provided for keeping the temperature of the entering water high. Here again a non-corrosive steel is desired which, if available, might encourage the further development of the integral or unit economizer for practically all types of boilers.

These two requirements call for material of practically the same physical characteristics. There is still another need for an improved non-corrosive steel in power plants. The last rows of the low-pressure sections of large steam turbines are very subject to corrosion. In many cases this rusting appears to occur only when the turbine is standing still. The moisture in the turbine condenses on the blading, and as air leaks in at the glands, oxygen goes into solution in the condensed steam and corrosion begins. Nickel steels

have been developed which are highly non-corrosive, but do not possess the required physical properties. Twenty-five per cent nickel steel fails very quickly from fatigue due to vibration and hence cannot be used, though a good resister of corrosion. Since these low-pressure blades are subjected to higher stresses than any others in the turbine, the required material must have a high tensile strength and must withstand shock and repeated vibrations, as well as have the property to resist corrosion.

It is thus apparent that the power-plant engineer must look to the metallurgist to provide him with new alloy steels to meet these recently developed requirements, and also that the need for such new steels is urgent.

Engineering Qualifications

NEW YORK papers recently devoted more or less space to an alleged interview with one Thomas Alva Edison, formerly of Menlo Park, in which, as a result of his "personally hiring and firing" aspirants for positions on the Edisonian staff, he is reported to have found college men "dull" and "amazingly ignorant."

If Mr. Edison is correctly quoted, it behooves our colleges (and it is to be assumed that technical colleges are particularly meant) to look sharply to their curriculums and pedagogical personnel or they will retrograde from that high estate which it is quite generally understood they have reached during late years as shapers of youthful mentality.

However, it may be well in the first place to apply the proverbial "grain of salt." It is darkly hinted that some time past, even since Parisians were startled by seeing one of their papers appear with a three-inch scarehead on the front page (for the first time in its history) reading "The Wizard is Here"—even since then it is hinted that he has indulged more or less in the habit of being interviewed.

And so, perhaps, an enterprising reporter did not catch Mr. Edison's exact words, for "the old man," as he is affectionately called by the pioneers and others, is sometimes hard to follow—very.

As to the basis of the alleged dictum, however, it is interesting to note some of the points covered by the questionnaire which, according to the story, was presented to candidates. For instance: What is copra? Where is Magdalena Bay? What is zinc? How did Cleopatra die? What are felt hats made of? Who founded the *New York Herald*? Perhaps there is a clue in this last question.

Admittedly, the more "all-around" education an engineer (especially a budding one) has, the better man and better citizen he will make, but is a knowledge of Cleopatra's history essential to defining a wattless current, or is familiarity with felt hats necessary in making a boiler test?

It is also naïvely stated that there were "mathematical and scientific questions put to the candidates," and perhaps these offered more familiar ground to the applicants for engineering jobs.

Somehow the report of the West Orange examination doesn't shake our belief that a collegiate or technical-school education is a fairly good asset, even if some of the best-known engineers in the country never possessed a sheepskin.

The Gas Producer

WHILE it is to be hoped that the high fuel prices of last year will not again worry the power plant, it is doubtful if the pre-war prices will ever prevail save for limited periods of low demand. Fuel has been one of the raw products of the country that has been wastefully consumed. Conservation methods of handling the situation as to other raw materials have been generally adopted, and the wasteful consumption of coal likewise must be checked. Any and all methods whereby a pound of coal can be made to do the work previously requiring two pounds is deserving of consideration.

Considerable discussion has been devoted of late to the gas producer. It is unfortunate that so many engineers are prone to hark back ten years and rehash the troubles that beset the early design of producer. The strongest advocate of the gas producer will not deny that errors were made in the first attempts to build a producer to handle American bituminous coals. The most radical opponent will likewise acknowledge that the early steam engine left much to be desired and that machinery of any kind seldom proves satisfactory when first built. The great trouble seems to be that the engineering public fails to realize that great strides have been taken in producer design, and it is distinctly unfair to attempt to gage present developments by the results of the pioneer producers.

It is not unusual to hear some fairly informed engineer state that the gas-producer industry was at the peak of its development in 1912, and that since then the use of producers has steadily declined. Investigation will reveal that the reverse is true. While its use was somewhat curtailed for a few years, at the present time there are many more producers in use than ever before. The majority are used for the production of industrial or furnace gas, but some three hundred are engaged in power-plant work. This number is considerably larger than in past years.

The greatest development to be expected is in the adoption of the gas producer in states where coal prices are high and where lignite is obtainable at a low cost. The Northwest and Southwest contain large lignite deposits that are of little use save in gas production.

In Australia there has been developed a producer that burns almost any kind of trash, such as wood, log bark, etc. Large quantities of such refuse are wasted in this country every year, and the saving to be secured by burning this material in a producer is worth investigation.

Suppose, Mr. Engineer, you were asked to make things more comfortable for the occupants of the Inspection Department, where the psychrometer shows a dry-bulb temperature of 74 deg., and a wet-bulb of 52 deg., and the windows cannot be opened because the tissue paper and other light packing stuff would blow away, could you approach this subject with calm equanimity? In any case you will be interested in an article shortly to be published in *Power*, by Charles L. Hubbard, a Boston engineer, on testing and improving ventilation in industrial plants.

The Blue Danube of musical fame is to be ignominiously put to work, according to the *Compressed Air Magazine*, which reports that a million-horsepower hydraulic power plant on the famous river above Vienna is being considered by the Austrian government.

CORRESPONDENCE

Railroad Stationary Engineering

Instead of "Railroad Stationary Engineering," Mr. A. D. Palmer's article, page 667 of the April 26 issue, would be somewhat properly titled "Poor Workmen in Railroad Shop." No doubt the abrogation of the National Agreement will do some good for the railroads, but it does not suggest that it can be blamed for the inefficiency that exists on some roads. It has been the hobby of many to cry about the railroads being handed over to the men. I believe that anybody closely associated with railroad life will disagree with this. Of course, in some places where the officers are not big enough to adjust themselves to the new conditions, such is the case, but these are very rare.

From my own experience in the last two years, after taking a position with an up-to-date railroad (coming directly from an industrial plant), I found the very same methods and rulings as described by Mr. Palmer. It was mighty hard to adjust myself to the new surroundings, which accompany every change in employment. After studying out every rule, after investigating every new method that suits railroad work, and being guided thereby, I encountered very little trouble.

In an industrial plant all power-house work is done by the employees in the power house. On a railroad all work, no matter where situated, is divided among the various crafts. The National Agreement boilermaker special rule No. 81 gives all boiler washing to the boilermaker helper, all stoker repair work to the machinist, boilermaker work to the boilermaker, and carpentry work, etc., to the carpenter. Sometimes it is necessary to consult the men to establish a rule for a certain class of work, but when once established and adhered to by the supervisor, it will bring no cause for complaint.

There have been many complaints about retaining in employment poor mechanics. I do not see where this blame rests on the men. The National Agreement gives a supervisor 30 days to find out if a man is suited for the job; if he is not, place him where he will be suited or discharge him. In three days I can usually find out what a man can do. After a man has been on a job 30 days, he cannot be discharged without a hearing. This is a good rule because it protects the employee from some supervisors who would discharge a man because of some personal feeling or prejudice.

The rule that allows a man five hours pay for three hours and twenty minutes or less is unjust more or less. I believe the majority agree on this. It is right that I should do an hour's work for an hour's pay.

As far as a supervisor not having authority to employ a man, I do not see where there is anything set forth in any rules or any other agreement that would prohibit him from so doing. The rule states that all jobs shall be bulletined for five days. This is necessary to let the eldest employee declare his seniority. All shops and factories to my knowledge consider their oldest workmen their best mechanics, with a few exceptions. If the oldest man is not capable, then no doubt he would not bid on the job. Someone younger may take it; if not, the foreman can assign a man. If a man cannot fill the job, there is nothing that can hold him on if the supervisor wishes to discharge him. When a hearing is held in a case of this kind, the odds are not in favor of the employee.

In writing this, I look at both sides of the case; it seems as though Mr. Palmer looks only at one side.

In conclusion I would state that boilermakers receive 85 cents an hour, not $1.08, and helpers 62 cents, not 93 cents an hour.

J. F. COEN.

Covington, Ky.

Efficient Burning of Fuel Oil

I wonder if C. W. Peters, who wrote regarding efficient burning of fuel oil, page 598, April 12 issue of *Power*, realizes that there are very competent fuel-oil engineers who do make installations, using not only some of the equipment he suggests, such as fuel-oil meters, recording thermometers and CO_2 machines, but also automatic damper controls and oil and steam regulators.

Many firms sell fuel-oil burning equipment like hardware, leaving it to the purchaser to make his own installation. No doubt Mr. Peters had this in mind. An ideal fuel-oil burning installation is made by a firm of fuel-oil engineers who take over the plant and install equipment that is suitable for that particular plant and so automatic as to be almost foolproof.

I know of installations in which regulators actuated by the steam working pressure in the boilers automatically control the supply of oil and steam to the burners. In conjunction and interconnected with these regulators are damper controls that have been set to regulate the amount of air passing through the furnace to obtain the maximum CO_2 possible.

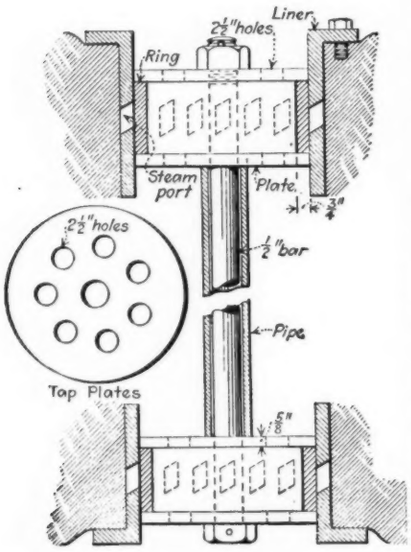
Those contemplating the burning of fuel oil often balk at the cost and endeavor to make the installation themselves, buying equipment in the open market. If they would seek the advice of competent fuel-oil engineers and allow them to make the installation, the saving in the fuel bill alone would warrant the small additional expense.

W. A. P. HENNINGER.

New York City.

Emergency Marine-Engine Repair

While on a voyage from Halifax to Barbados, Trinidad and Demerara, and when 250 miles north of Barbados, the nut of the intermediate-pressure piston rod at the crosshead end came off. The piston, being forced up, fractured the cylinder cover and cylinder walls, making the engine useless. We first blocked with wooden wedges the bottom steam ports admitting steam to that cylinder and filled the cylinder with about fifteen inches of cement, fireclay and bricks. Wooden wedges were also driven in the top steam ports. The piston valve was then put in and the top ring raised off the flange one inch, so as to allow the steam from the high-pressure cylinder to flow into the exhaust leading to the low-pressure cylinder. Steam is admitted to the intermediate cylinder with an outside lead and exhausts at the center of the valve. The connecting-rod brasses were disconnected from the crankshaft, also the eccentric straps, to allow everything for the intermediate cylinder to be free.



HOW THE STEAM PORTS WERE PLUGGED

We proceeded to Barbados and got a seaworthy certificate from Lloyd's surveyor and then we proceed to Trinidad and to Demerara, where more permanent repairs could be made to take us to a loading port and home. In Demerara we fitted wooden pieces tightly into the valve stem ports, top and bottom, and turned two rings ¾ in. thick to make a tight fit in the valve liners so as to close the steam port. Two plates having 2½-in. holes in them were placed at both sides of the rings, making a snug fit in the valve liners. A long rod was run through the center of these plates and a nut fitted at both ends. A pipe in the center between the top and the bottom plate was used for a distance piece. When the nuts were screwed hard up they kept the rings and plates in the same place (see illustration).

We carried 180 lb. pressure and were getting too much pressure on the low-pressure cylinder. Of course we did not continue that pressure, but reduced it to 130 lb., which with the low-pressure expansion linked up to 20 lb. developed 262 i.hp. in the high-pressure cylinder and 350 in the low-pressure cylinder.

Then instead of shifting the eccentric sheave of the low-pressure cylinder we pinned a ½-in. piece of plate on the top and bottom of the slide valve. This allowed us to open up more on the high-pressure cylinder. Of course the pressure on the low-pressure slide-valve chest was more, but the power was the same, and we increased the high-pressure cylinder to 300 and the low-pressure cylinder was about 350 i.hp., made more revolutions and gained in speed owing to increased power.

We left Demerara for Barbados and then proceeded to Nuevitas, Cuba, loaded a full cargo of sugar for Philadelphia, discharged there and proceeded to Halifax, light, where we are now waiting for a new cylinder, having traveled 4,440 miles and encountered all sorts of weather. The engine was making the same revolutions with the two as was previously made with the three cylinders.

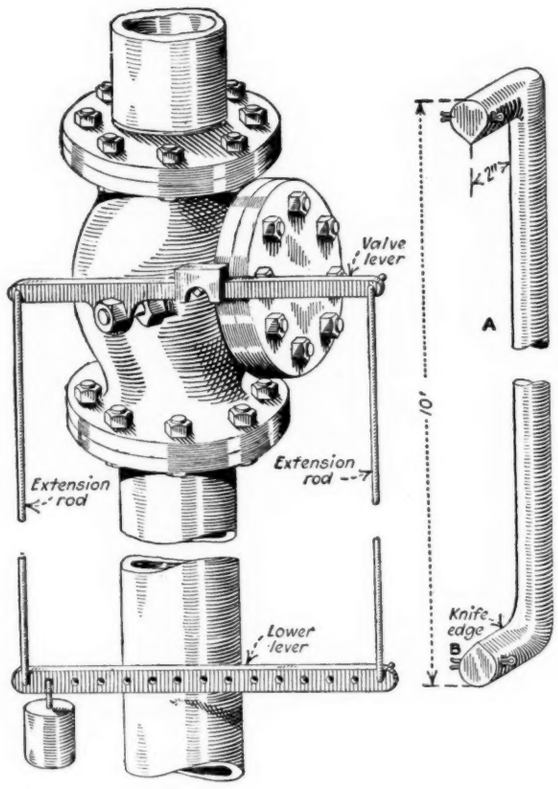
GEORGE STEPHEN.

Halifax, Canada.

Device for Adjusting Back-Pressure Valve

The back-pressure valve in a 16-in. vertical exhaust pipe leading to the atmosphere from a power plant is located about 15 ft. above the floor, and the intervening space is filled with a network of small pipes. This prevents placing a ladder so that the valve can be approached conveniently for adjustment.

The engineer solved the difficulty with the device shown in the illustration. A ¾-in. hole was drilled in each end of the weight lever which is attached to the valve. The ends of two ½-in. rods, 10 ft. long,



LONG-RANGE ADJUSTING DEVICE FOR BACK-PRESSURE VALVE

were then bent at right angles, as shown at *A*, and were filed to knife-edges as shown at *B*. The right-angled ends were also drilled for the insertion of ⅛-in. cotter pins. The upper ends of the rods were hooked into the holes in the ends of the valve lever, while the lower ends were hooked into holes in the ends of another lever having the same length and cross-sectional area as the valve lever.

A series of ¼-in. holes were drilled in the lower lever, the distance between successive holes increasing by increments of ¼ in. That is, the distance from the

first hole at the left to the second hole is 3 in. The distance from the second hole to the third is 3¼ in., and so on through. The holes in the bottom lever are not shown to scale.

Before attaching the extension rigging, the engineer set the weight on the valve lever for a back pressure of about 1½ lb. But the supplementary weight attached to the lower lever is such that when it is hung from the hole at the left-hand end, the back pressure is about ¼ lb. When it is hung from the second hole from the left, the back pressure is about ½ lb. Thus the back pressure increases by increments as the supplementary weight is shifted from hole to hole toward the right. When the supplementary weight is suspended from the hole at the right-hand end, the back pressure is about six pounds.

A. J. DIXON.

St. Louis, Mo.

Using an Indicator on an Oil Engine

Many Diesel-engine stations do not posses an indicator and the valve settings are never checked by indicator diagrams. The writer has charge of a Diesel-engine plant in a cotton mill, and his experience proves that an indicator is almost a necessity.

Originally, the engine was run without any indicator diagrams being taken. In Fig. 1 is shown a diagram after six months' operation. The sharp point to the combustion line at *a* was at first assumed to be a result of early fuel admission. The timing was made later and later, each adjustment causing the engine to smoke

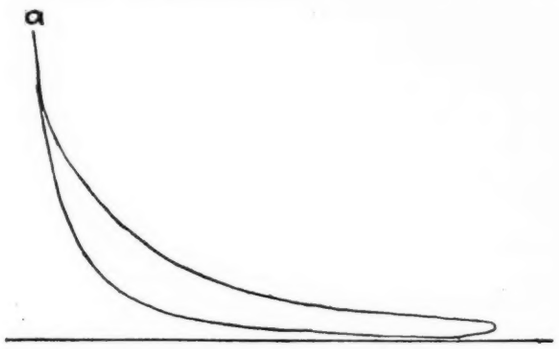


FIG. 1. INDICATOR DIAGRAM WITH LATE INJECTION

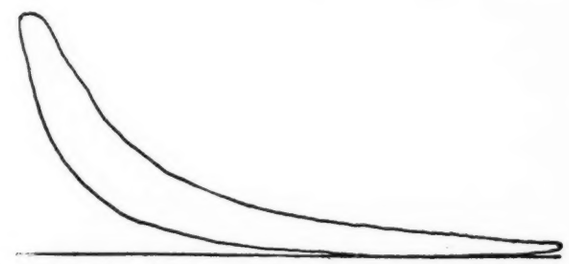


FIG. 2. DIAGRAM WITH 487 LB. COMPRESSION

more heavily. Finally, the timing was made early and the smoke was eliminated. In checking the valve timing, it was found that the camshaft had considerable play and, when running, gave entirely too much clearance to the fuel-valve rocker arm. Without the indicator this would not have been detected.

In experimenting with various compression pressures we found that the compression of 44 atmospheres shown by the diagram in Fig. 1 was entirely too high for the best economy. Regardless of the timing of the fuel valve the combusion line never became flat.

As a result of a decrease in the compressor pressure the diagram appearing in Fig. 2 was obtained. This diagram, showing 32½ atmospheres of compression pressure or 487 lb. per sq.in., reduces the working stresses

FIG. 3. COMPRESSION DIAGRAM

on the engine. The decrease in frictional loss produced a saving of 13 to 16 per cent in fuel.

Fig. 3 is a diagram taken with the blast air and fuel lines shut off. The air in the cylinder is compressed to a final pressure at *a* along the line *abc*. If no heat was lost to the cooling water, *adc* and *abc* would coincide. There being some loss, *adc* shows lower pressure at each given volume. If the difference is great, it is evidence that the piston is leaking air past it.

It is the writer's opinion that every engineer should obtain an indicator and use it. No better way is open toward the advancement of the Diesel engineer than in increased operating efficiency. This is more easily secured if the proper instruments are at hand.

Ahmedabad, India. P. P. TALATY.

Operating Diesel Engines

Almost every day some stranger drops into our Diesel engine plant. Among the questions hurled at us are: "Do you break many cylinder heads?" "How about crosshead pin troubles?" "Broke any crankshafts yet?" When we answer, "No," they say, "Gee, that's funny, I always thought Diesel engines gave trouble."

I always tell them that if engineers used "horse sense" and were on the job, no trouble would occur. If the chief knows his business and trains his men, operating a Diesel plant is by no means as hard as a steam plant. Any engine, whether it be Diesel, surface-ignition, gas or steam requires attention, and the troubles that have occurred with Diesels are due to the operator not knowing his job.

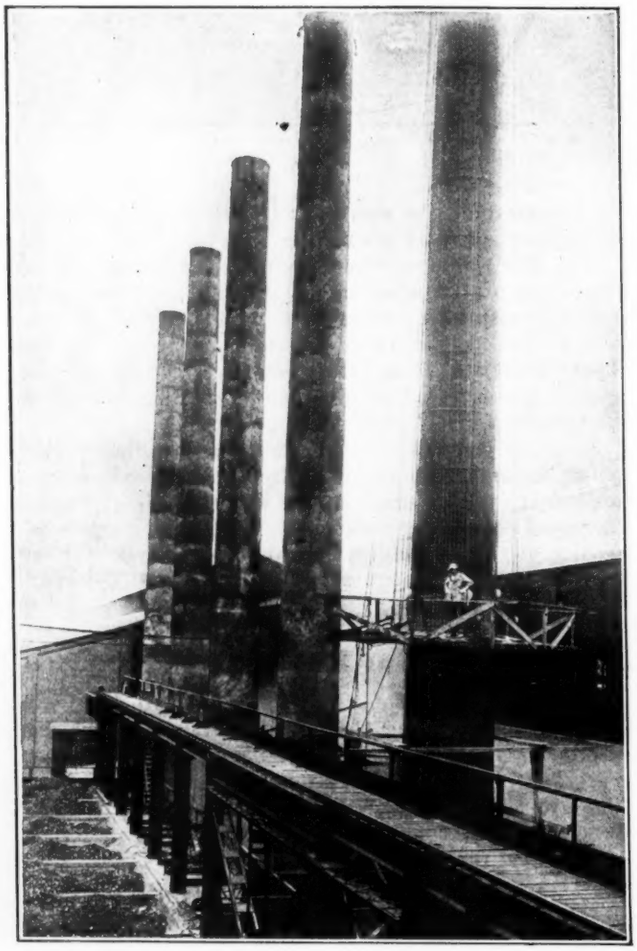
We have two 200-hp. Dow Diesels which have been operating continuously for six years, 24 hours a day, with but one three-week shutdown each year to overhaul and check the bearings. In fact, we have run one engine 72 days without a stop. During the last six years we have renewed four crankpin brasses, two main bearing shells and sixteen exhaust valves. The main bearing was damaged, owing to a stuck oil ring and an inexperienced operator. The crankpin brasses were lost because of dirty oil.

I have been running Diesel engines for eleven years and find that they give little trouble when the engine-room force is properly organized and trained. Above all the Diesel engineer must be a good mechanic.

San Francisco, Cal. JAMES R. PETRIE.

Incasing Steel Stacks with Concrete

In the March 8 issue of *Power* illustrations were presented showing the demolition of some concrete-incased steel stacks, with a short description of the method employed in dropping these stacks. The demolition of these stacks was undoubtedly justified by their condition, and the work was cleverly planned and executed. Upon reading the description and looking at the illustrations, however, readers generally may decide that the incasement in concrete of a steel stack that has corroded to the danger point is of questionable value and likely to result in disappointment and waste of money. Such a conclusion would be erroneous. A careful investigation discloses that there were good reasons why these stacks failed. It is not the purpose of this article to discuss these reasons, but simply to say that the conditions were abnormal.



FOUR STEEL STACKS INCASED WITH CONCRETE, FIFTH BEING CEMENTED

The conservation of self-supporting steel stacks that have corroded to the danger point, by incasing them in concrete, is becoming an established practice. To demolish an otherwise serviceable stack simply because the steel shell has lost its strength by corrosion, when at comparatively small expense and without shutting down the boilers, it may be converted into a permanent concrete stack, would, in the light of present knowledge, be a mistake. In the accompanying table is given a list of steel stacks that had lost their strength by corrosion and have been converted into permanent concrete stacks.

In reference No. 1 of the table, two of the five stacks were out of commission when incased and hence cold. The other three were in regular service and at normal temperature. Stack No. 2 was at a pumping plant that could not be shut down. The work was done in July without interrupting service.

STEEL STACKS INCASED WITH CONCRETE

No.	No. of Stacks	Size Diam., In.	Height, Ft.	Year
1	5	90	125	1918
2	1	115	175	1918
3	2	68	110	1918
4	2	60	100	1920
5	3	68	110	1920
6	6	90	150	1921

The two stacks of reference No. 3 were incased in December, when the weather was so cold that the work had to be protected against freezing. Both stacks were in regular service. Stacks No. 4 were done in July. One was in service and the other out of commission.

The three stacks of No. 5 were incased in December, and work is now in progress on the six stacks of reference No. 6. The first one was done in January when weather was cold and the stack out of commission. Sufficient fire was made in the stack to bring the steel plate above the freezing point. The other three were in commission and the remaining two will be done while in service.

In the table the list comprises nineteen stacks, including two not yet finished. Seventeen of these are in daily use and eight of them have gone through two winters. None of them shows any defects, except a few surface craze cracks, which do not penetrate perhaps more than one-eighth inch.

This is to be noted, however. Every one of these steel stacks was lined with firebrick to the top. This provided from 4½ to 9 in. of heat insulation and an air space of at least one inch between the firebrick and the steel shell, so that the temperature of the latter probably never exceeded 150 deg. Any fluctuation in temperature of this steel would take place gradually, and the surrounding concrete could follow in this fluctuation. The steel shell and the concrete undoubtedly contracted and expanded together, and any stress in the concrete that may have been due to unequal contraction and expansion was distributed by the substantial horizontal reinforcement that was provided. All the stacks were built up with the cement gun.

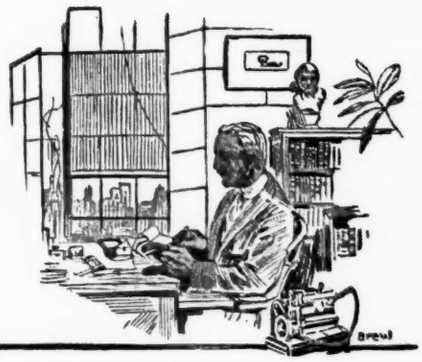
The five stacks of reference No. 1 are shown in the accompanying photograph, four of them having been completed, while work on the fifth is in progress. The heavy reinforcing steel is visible.

It is, then, evident that the incasing of an otherwise serviceable self-supporting steel stack properly lined, is entirely practical, and the work can be done without interruption of service. From past experience, the new stack will not crack, it will not need painting and will resist corrosive gases. If any portion of the original steel stack corrodes from within, it will not matter, as the stack does not depend on the original steel shell for its stability or permanence.

JOHN V. SCHAEFER, President
Chicago, Ill. Cement-Gun Construction Co.

Of the gas and oil engines now being imported by Argentina about 90 per cent are from the United States, instead of 50 per cent, as before the war.

INQUIRIES OF GENERAL INTEREST



Waste Heat Recoverable from Diesel Engine—How much heat can be obtained from an exhaust heater on an oil engine? G. N.

If part of the jacket cooling water is sent through an exhaust heater, over 35 per cent of the heat in the fuel supplied to the engine can be recovered with a final water temperature of 212 degrees.

Voltage Regulations in Over-Compounded Generators—In an over-compounded direct-current generator why is the voltage rise from no load to full load different at different speeds, assuming the speed to remain constant from no-load to full load in each case, and the no-load voltage to be the same in all cases? J. H. B.

In an over-compounded direct-current generator the voltage at constant speed rises from no load to full load because the series field adds a certain amount of flux to the no-load flux, which adds a certain voltage to the no-load voltage. If the speed is lowered, the series-field flux is cut at a lower rate by the armature conductors, and less voltage will be added to the no-load voltage.

Placing Storage Batteries Out of Commission—If it is desired to take a storage battery out of service and let it remain idle for a long time, should it be given a final charge or completely discharged before removing the elements from the jars? R. N. P.

Before taking a storage battery out of service, even for a short time, it should be given a final charge. Under no conditions should the plates be removed when the battery is discharged, and allowed to dry, for their surfaces are covered with lead sulphate that hardens, cracks and flakes off when dry, resulting in loss of active material. If the battery has been previously charged the sponge lead and lead peroxide that compose the plates have much less tendency to flake off when dry.

Valves and Fittings for Superheated Steam—Why is it difficult to obtain durability of valves and fittings used with superheated steam when made of material whose strength is not reduced when heated to the same temperature by ordinary methods? S. V.

In many cases where valves and fittings are used with superheated steam there is continual "surging" of the steam temperature over a considerable range and failure of the material probably is due to separation of particles like check cracks which are formed in some metals when suddenly quenched from high temperatures. Frequent changes of temperature are also destructive of the material from inequality of expansion, which gives rise to change of form with repeated "hingeing" of one part of the valve or fitting on another part of the same piece of metal.

Efficiency Required for Riveted Joint—What efficiency of longitudinal riveted joint would be required for a boiler 72 in. in diameter, made of ½-in. plate of 55,000 lb. tensile strength, to carry 125 lb. working pressure with a factor of safety of 5? C. J. Y.

A boiler 72 in. in diameter under a working pressure of 125 lb. per sq.in. would be stressed across a longitudinal joint $(72 \times 125) \div 2 = 4{,}500$ lb. per inch length of the joint, and with a factor of safety of 5 the required strength per inch length of the joint would be $4{,}500 \times 5 = 22{,}500$ lb. If the tensile strength of the plate material is 55,000 lb. per square inch of cross-sectional area, then for ½-in. plate the hoop strength of the solid plate per inch length of the shell would be $1 \times \frac{1}{2} \times 55{,}000 = 27{,}500$ lb. and the efficiency required for the joint would be $22{,}500 \times 100 \div 27{,}500 = 81.8$ per cent.

Starting and Priming Pipes for Pumps—What is the advantage of using starting and priming pipes for a reciprocating pump? R. N. L.

A starting pipe assists in starting by allowing the air to escape and relieving the force chamber of all pressure until the capacity has increased sufficiently to open the check valve, when the starting valve may be closed. Upon starting with a suction lift with the pump and suction pipe empty of water, the pump will operate as an air pump, the air is imperfectly removed from the clearance spaces and the pump may remain airbound for a considerable time. When the pump chambers are filled with water the clearance spaces are made harmless and the pump picks up promptly. If the suction pipe is fitted with a foot valve it usually is sufficient to prime only the suction pipe and the pump chambers will fill and the air will be driven out through the starting pipe. But if there is no foot valve each pump chamber must be separately supplied from the priming pipe.

Setting Link-Motion Valve Gear—What is the method of setting the eccentrics and valve of a link-motion valve gear? A. I.

Assuming that the link motion is designed for equal leads, place the link at full gear forward and, with the crank on a dead point, set each eccentric as near as may be with the proper angle of advance, remembering that the centers of the eccentrics should be on opposite sides of and at the same angle (more than 45 degrees) with the crank. Give the valve the proper lead by adjusting the length of the valve rod or forward eccentric rod. Then turn the engine into the other dead point, and, if the valve does not give the proper lead, change the length of the eccentric rod or valve rod to correct half of the error and shift the forward eccentric until the proper lead is obtained. Then place the link at full gear backing, and in the same manner as before obtain the proper lead with the crank on one center and then the other. Again place the link at full gear forward and see if the setting has been disturbed. If it has the valve must be reset by the same method until equal loads are obtained. Usually a single repetition of the adjustments in forward and backing gear will be found sufficient. If the motion is designed for unequal leads, as commonly the case with marine engines, the processes will be the same, only in that the adjustments must be made to the proper lead for each end.

[Correspondents sending us inquiries should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications and for the inquiries to receive attention.—Editor.]

Interpretations of the A. S. M. E. Boiler Code

REQUESTS for interpretations of the Boiler Code are acted upon by the Boiler Code Committee, their formulated replies being submitted to the Council of the American Society of Mechanical Engineers. The inquiries and replies are published from time to time in *Power*, with such explanatory comments and illustrations as seem advisable for a clear understanding of the interpretations on the part of engineers in general. The last group of interpretations published in *Power* was in the issue of April 12, 1921.

Case No. 322 (Reopened)—Inquiry: What material is it necessary to use under the requirements of the Boiler Code for the Y-fitting, also for the safety-valve body, for safety valves to be operated at a pressure of 225 lb.? May cast iron be used for this purpose?

Reply: It is the opinion of the committee that the Y-fitting referred to may be made of cast iron provided the temperature does not exceed 450 deg. F. as stipulated in Par. 12. The safety-valve body need not be of steel unless it is to operate with superheated steam. (See Par. 289)

Par. 12 provides that cast iron shall not be used for nozzles or flanges attached directly to the boiler for any pressure or temperature, nor for boiler and superheater mountings, such as connecting pipes, fittings, valves and their bonnets, for steam temperatures of over 450 deg. F.

Par. 289 provides that every safety valve used on a superheater discharging superheated steam shall have a steel body with a flanged inlet connection, and shall have the seat and disc of nickel composition or equivalent material, and the spring fully exposed outside of the valve casing so that it shall be protected from contact with the escaping steam.

The interpretation given in Case No. 344 (see elsewhere on this page) may be of interest in this connection.

Case No. 337—Inquiry: In applying reinforcing plates within the drums of water-tube boilers to strengthen the shell where the tubes enter, as shown in Fig. 1, what is the requirement of the Boiler Code for the thickness of the plate and the strength of the riveting to compensate for the material removed in forming the tube holes?

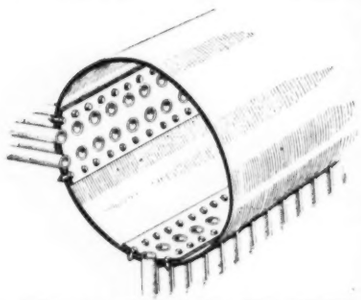


FIG. 1. (CASE NO. 337) REINFORCING PLATES IN DRUM

Reply: There is no rule in the Code for the design of reinforcing plates to be used for this purpose. The committee advises against such reinforcing plates; but if they are to be used, the committee is of the opinion that if the thickness of the plate be made sufficient to give, when added to the thickness of the drum shell, a ligament efficiency under Par. 192 or 193, at least equal to that of the longitudinal joint of the shell, it will then only be necessary to so rivet the plate to the shell that the joint will calculate to an efficiency also equal to the longitudinal joint of the shell.

The method of construction referred to is shown in Fig. 1, which illustrates the relation between reinforcing plates, tubes, and shell of drum.

Case No. 342—Inquiry: Is it permissible in a heating boiler to be operated at a pressure not to exceed 15 lb. per sq.in., where cross-bracing is required in the side-furnace sheet of a locomotive-type boiler between the center line of the cylindrical portion and the crown sheet, to locate a through cross rod one-half of the staybolt pitch below the center line, as shown in Fig. 2?

Reply: It is the opinion of the committee that the method of staying described does not conflict with the requirements of the Code when the working pressure does not exceed 15 lb. per sq.in.

To be theoretically correct, the staying in this case (see Fig. 2) should obviously begin at the center line of the cylindrical portion of the shell, or at the point of juncture between the cylindrical and flat portions. Structurally, however, it is evidently preferable to locate the line of through rods as suggested in the inquiry and shown in Fig. 2, and in the opinion of the committee the tendency to distortion as indicated by the dotted line would be negligible for the low maximum pressure specified.

Case No. 343—Inquiry: What constant will it be necessary to use in the formula in Par. 199 of the Boiler Code, for a surface braced with through stays having inside and outside nuts, but omitting washers?

Reply: It is the opinion of the committee that where through stays are used with inside and outside nuts, but omitting washers, the constant of 135 should be used in the application of the formula in Par. 199.

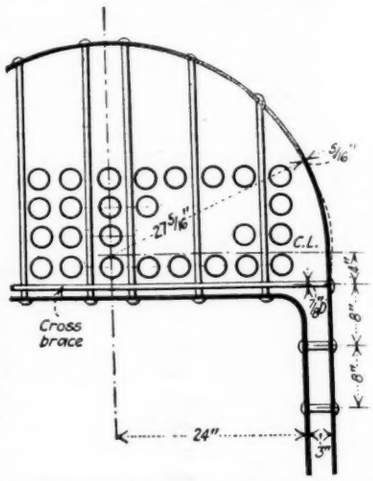


FIG. 2. (CASE NO. 342) ALLOWABLE LOCATION OF THROUGH CROSS ROD FOR 15 LB. MAXIMUM PRESSURE

Par. 199 provides that the maximum allowable working pressure for various thicknesses of braced and stayed flat plates and those which by these rules require staying as flat surfaces with braces or staybolts of uniform diameter symmetrically spaced, shall be calculated by the formula $P = C \times \frac{T^2}{p^2}$ in which P equals maximum allowable working pressure in pounds per square inch; T equals thickness of plate in sixteenths of an inch; p equals maximum pitch measured between straight lines passing through the centers of the staybolts in the different rows, which lines may be horizontal, vertical or inclined, in inches; and C a constant corresponding to different methods of fastening the ends of the stays. The constant of 135 given in the interpretation corresponds to that for stays screwed through plates and fitted with single nuts outside of plate.

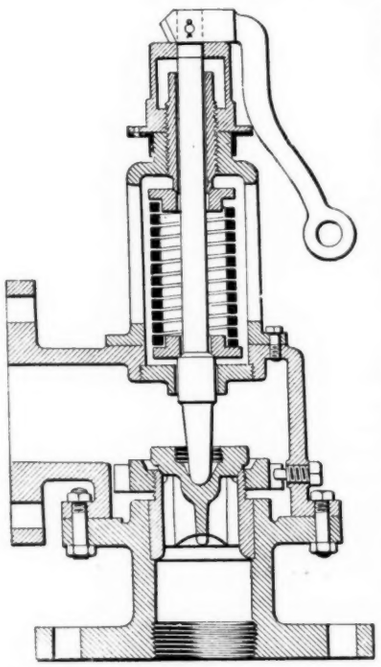


FIG. 3. (CASE NO. 344) TYPICAL SAFETY-VALVE DESIGN

Middle and lower parts constitute "body" and must be of cast steel.

Case No. 344—Inquiry: In complying with the requirements of Par. 289 for a safety valve used on a superheater, is it necessary that the entire casing of the safety valve be made of steel?

Reply: It is the opinion of the committee that the intent of Par. 289 is that the body of a safety valve includes all parts of the body and casing which come in contact with the boiler steam or the steam discharged, and therefore in the construction submitted the casing shall be made of cast steel for use with superheated steam.

Case No. 344. The design submitted to the committee and referred to in the preceding reply was similar to that shown in Fig. 3, except that it was in two parts, the middle and upper parts shown in the illustration forming one casting. In accordance with the ruling of the committee both the lower and middle portions of the design shown in Fig. 3 must be of cast steel, since they come in contact with the steam, thereby consti-

tuting the "body" of the valve, whereas the upper part or spring housing may be of cast iron.

Case No. 345 — Inquiry: Is it the intention of Par. 335*b* of the Boiler Code, that hot-water boilers must be built according to the rules for power boilers when either the grate area exceeds 10 sq.ft. or the maximum allowable working pressure exceeds 50 lb. per sq.in., or is it the intention that this requirement applies only when both of the above limits are exceeded?

Reply: It is the opinion of the committee that the requirement of Par. 335*b* applies only when both of the limits therein specified are exceeded in the hot-water boiler.

The requirements referred to read as follows: "The rules for power boilers shall apply . . . .*b*. To all steel plate *hot-water* boilers where the grate area exceeds 10 sq.ft. and the maximum allowable working pressure exceeds 50 lb. per sq.in."

Case No. 346—Inquiry: Is it permissible, in the application of the requirements in the A.S.M.E. Boiler Code for safety valves for heating boilers, to take advantage of the proportional difference between bevel-seated and flat-seated valves provided for in the safety-valve rules for power boilers?

Reply: It was the intention of the committee that Table 9 of the Boiler Code should cover the application of safety valves to all heating boilers, no matter what the character of the seat. Attention is, however, called to the fact that where the conditions specified in Table 9 are exceeded, the formula in Par. 358 may be used.

Par. 358 provides that when the conditions exceed those on which Table 9 is based the following formula for double and flat-seated valves shall be used: $A = \frac{W}{4.7P}$, in which A equals area of direct spring-loaded safety valve per square foot of grate surface in square inches; W equals weight of water evaporated per square foot of grate surface per hour in pounds; and P equals pressure (absolute) at which the safety valve is set to blow, in pounds per square inch.

Changes in Massachusetts Boiler Rules

The steam boiler rules of Massachusetts, as formulated by the Board of Boiler Rules, Department of Public Safety, are being reprinted, and at an executive session of the board, following a public hearing in the State House, Boston, on May 5, 1921, the following changes in the rules as previously published under date of Jan. 3, 1919, were adopted.

Page 14, paragraph 2, section 2, part 2—(Add to Condition C): Where two steam mains carrying different pressures are connected together a safety valve or valves installed on the connecting pipe and equal in area to the area of the pipe and set at the lower pressure allowed shall be considered to meet the requirements of the foregoing paragraph.

Page 45, paragraph 13, section 1, part 3—Strike out the word "heat" and make paragraph read: "Each plate shall be distinctly stamped by the manufacturer with the melt or slab number."

Page 138, paragraph 3, section 6, part 3—Boilers in state prior to May 1, 1908, may be reinstalled:

Boilers in this commonwealth prior to May 1, 1908, where both ownership and location are changed, which do not conform to the rules of construction formulated by the Board of Boiler Rules, may be installed after a thorough internal and external inspection and hydrostatic-pressure test by a boiler inspector of the division of inspection of the Department of Public Safety, or by an inspector holding a certificate of competency as an inspector of steam boilers for this commonwealth and employed by an authorized insurance company.

The pressure allowed on such boilers when reinstalled shall be ascertained by the use of the following factors of safety:

(a) Six for boilers the longitudinal joints of which are of lap-riveted construction, diameters up to and including thirty-six inches.

(b) Eight for boilers the longitudinal joints of which are of lap-riveted construction, diameters over thirty-six inches.

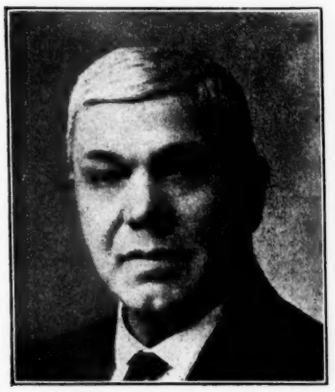
(c) Five for boilers the longitudinal joints of which are of butt and double-strap construction, age not exceeding ten years.

(d) Five and five-tenths for boilers the longitudinal joints of which are of butt and double-strap construction, age over ten years.

It was also decided to take steps soon looking toward the formulation of rules for low-pressure steam boilers.

The New Chief of the Calumet Station

Chief engineer of what is planned to be one of the largest and most modern power plants in the country is the latest honor tendered A. L. Hadin. The plant is the new Calumet Station of the Commonwealth Edison Co., of Chicago. Among engineers throughout the country Mr. Hadin is well known, and his record at the Fortieth Street south side elevated plant is one that has brought him unstinted praise. Here, with nine cross-compound Corliss engines driving railway generators of an aggregate capacity of 11,700 kw. and with 22 water-tube boilers having a total of 98,000 sq.ft. of steam-making surface, he maintained for 16 years an average operating cost of 0.42c. per kilowatt-hour. This included repairs, maintenance, and in fact everything but fixed charges.



A. L. HADIN

Since the age of nine Mr. Hadin has been identified with power-plant work. In his early days in Sweden he became engineer on a steamboat, and after coming to this country in 1880 he took a position as chief engineer in a flour mill at Clay Center, Kan. Five years later he was given charge of a privately owned water-works serving the city, which at that time had a population of 18,000. A year later the same firm, Fanstock & Co., of New York, transferred him to Hutchinson, Kan., to superintend the water-works, electric-light station and the gas works. Here he remained until 1891, when he was again transferred to Aurora, Ill., to become chief engineer and master mechanic of the street-railway company. He remained for six years, and on Dec. 1, 1897, his services were secured by the South Side Elevated Railroad, Chicago. At that time the elevated railway was electrified and the power plant built at 40th and State Sts. With an initial installation of four 800-kw. units and with other units of 1,500 and 2,000 kw. capacity added from time to time until the aggregate capacity given was reached, Mr. Hadin made the remarkable record previously mentioned.

When the elevated roads of Chicago were consolidated in 1911 he was made chief operating engineer for the system. His good work continued, and in the Union Loop and Metropolitan stations savings netting respectively $4,000 and $8,000 per month were effected.

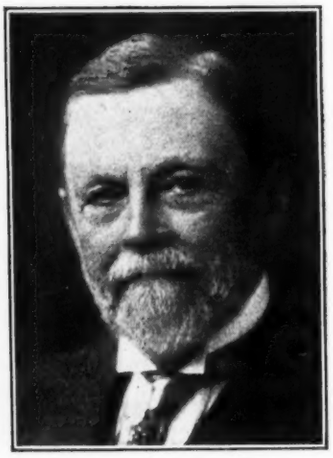
In 1913 all these plants were taken over by the Commonwealth Edison Co. and Mr. Hadin became chief engineer of railway power-houses. For eight years he has served in this capacity. His ability had long been appreciated, and finally, on April 18, the latest recognition came.

Western Engineers Choose Officers

Charles H. MacDowell was elected president of the Western Society of Engineers at the election held on May 4. Mr. MacDowell is president of the Armour Fertilizer Works and brings to this position a wide experience as an executive. The other officers elected are: First vice-president, Julius L. Hecht, assistant to vice-president, Public Service Co. of Northern Illinois; second vice-president, F. F. Fowle, consulting electrical and mechanical engineer; third vice-president, Benjamin B. Shapiro, consulting structural engineer; treasurer, Homer E. Niesz, manager of industrial relations, Commonwealth Edison Co.; trustee, Edwin W. Allen, assistant district manager and district engineer, General Electric Co. The members of the Washington Award Commission are: John F. Hayford, director School of Engineering, Northwestern University, and Henry J. Burt, manager, Holabird & Roche. The officers-elect will take office on the first Wednesday in June.

Charles H. Morse Dies

Charles Hosmer Morse, the founder, chairman of the board of directors, and for many years the president of Fairbanks, Morse & Co., died on May 5 at his home in Winter Park, Florida, after an illness of several weeks' duration. Mr. Morse was born in St. Johnsbury, Vt., on Sept. 23, 1833, and as a boy entered the employ of E. & T. Fairbanks & Co., the scale makers, whose factory was located in his home town. In 1857 he was sent to Chicago to assist in the establishment of a branch house there for E. & T. Fairbanks & Co., and after a few years went to Cincinnati to take charge of the firm's business in that territory. In 1870 the development of the business in Chicago necessitated his return there, and shortly after this the present firm of Fairbanks, Morse & Co. was formed to handle the Western business of the scale company.



CHARLES H. MORSE

In 1883 Mr. Morse acquired control of the Eclipse Wind Engine Co., at Beloit, Wis., and under his guidance the concern developed into the largest windmill manufacturing concern in the world. At about this same time the interest of W. H. Williams, inventor of the Williams steam engine, was acquired and this line added.

From a modest beginning as Western sales agent for Fairbanks scales, Mr. Morse gradually acquired other lines until at the present time a broad line of power and weighing machinery adapted to use in every industry is manufactured and sold by Fairbanks-Morse.

In the field of oil-engine development Mr. Morse was a pioneer. Early in the 90's, in conjunction with James A. Charter, inventor of a gasoline engine, experimental work was begun and the result was the complete line of oil engines now made by the company. Besides this work in developing the oil engine to its present state of practicability, Mr. Morse has also done notable work in the electrical and railway appliance field. The development of a line of alternating-current and direct-current ball-bearing motors and the perfection of railway motor cars, from a small beginning to their present state of almost universal use, are attributable to his foresight and energy.

Mr. Morse is survived by his widow, Helen H. Morse, two sons, Charles Hosmer and Robert Hosmer, who have been associated with him in business for many years, and one daughter, Mrs. Richard M. Genius.

Great Variety of Papers in Store for the N. E. L. A. Convention

The tentative program is now at hand for the convention of the National Electric Light Association, which is to be held from May 31 to June 3 at the new Drake Hotel, Chicago. A great number of reports, addresses and papers is scheduled, so that those who attend the convention will find their time very fully occupied.

The convention will be opened at 9:30 Tuesday morning, May 31, by an address of welcome to Chicago, followed by a speech from Martin J. Insull, president of the association. Later on in the morning addresses will be made by J. E. Davidson, chairman of the public relations national section, and by I. E. Moultrop, chairman of the technical national section. At 2:15 in the afternoon of the same day (May 31) the technical national section will meet for an address by the chairman and for reports from the electrical apparatus committee and the safety rules committee by their respective chairmen, R. F. Schuchardt and Thomas Sproul. Simultaneously, the commercial national section will meet to receive reports on "Economics of the Use of Central Station Electric Power," from J. W. Meyer, and on "General Power," from George H. Jones.

In a general session Wednesday morning, June 1, President Insull will make an address on the general subject of "Power," and at about noon a report on the Superpower Survey will be made by Charles L. Edgar. In the afternoon the technical national section will meet and reports will be made by the committees on overhead systems and on inductive interference. During the same afternoon a paper entitled "Accounting for Coal," will be given by H. M. Edwards, of the New York Edison Co., in the accounting national section meeting, and in the public relations national section meeting a report will be presented by the committee on relations with regulatory bodies.

Thursday morning in the general and executive session the committee on electrical resources of the nation, headed by M. S. Sloan of the Brooklyn Edison Co., and the committee on water-power development, headed by Franklin F. Griffith, of the Portland Railway, Light and Power Co., will present their reports.

Thursday afternoon the bonus systems committee of the accounting national section will make its report, as will also the underground systems committee of the technical national section. Two important reports will be those of the hydraulic power committee and the prime movers committee, headed respectively by Markham Cheever, of the Utah Power and Light Co., Salt Lake City, and N. A. Carle, of the Public Service Electric Co., Newark, N. J. These two reports will be given at 2:30 and 4:00 p.m., Thursday, June 2, at the technical national section meeting.

Friday morning the committee on electrification of steam railroads will report at the general and executive session.

The features of the program mentioned here include only those of interest in the field of power. Many of them will have an important bearing on the electric-power industry. Further particulars of the convention and the complete program may be obtained from the chairman of the convention committee, E. W. Lloyd, Commonwealth Edison Co., 72 West Adams St., Chicago.

Tells Executives to Wake Up

L. W. Wallace, executive secretary of the American Engineering Council of the Federated American Engineering Societies, in a recent speech before the Society of Industrial Engineers made a strong appeal for that broader attitude of mind and increased responsibility on the part of industrial leaders that is being discussed now so frequently. Mr. Wallace said:

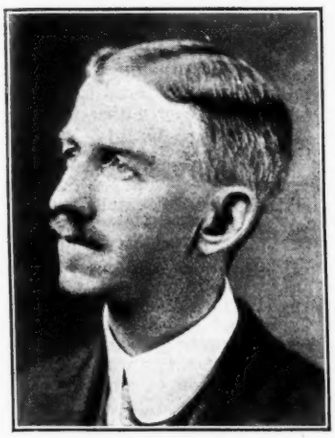
Today, as never before in the history of industry, there is a necessity for candor and absolute honesty. More information and less hearsay is the need of the hour. We can conceive of no more important tasks for the industrial leader to perform than just this: Give the economic facts governing world industry to the masses of industrial workers.

With such a background formed, then take them into your confidence as regards your business. Show them the books of the company, inform them of the financial condition of the company and as to the probable business for the year. If this be done, much hearsay and the restlessness and disturbance resulting therefrom will be eradicted.

The industrial leader needs to be more than purely an industrialist, just as Herbert Hoover, an engineer, is more than purely an engineer. Unfortunately, the daily schedule of too many industrial leaders reads as follows: Home to plant—plant to lunch—lunch to plant—plant to home. An eternal triangle, almost as deadening as that other eternal triangle that we so frequently hear of. The industrial leader, by virtue of the fact that he has within his control the destinies of hundreds or thousands of workers and their families, has a peculiarly significant responsibility to meet as a citizen. He cannot shut his eyes to the condition of the schools, of the streets, of the public utilities of the community in which his workers reside. If he fails to make his executive ability felt in all matters of community concern, then he has failed to meet his responsibility as an industrial leader.

New York's Boiler Inspector Dies

George A. O'Rourke, for more than seven years chief engineer of the Bureau of Boilers and Explosives of the New York State Department of Labor, died April 24 at his home in Albany and was buried in Brooklyn April 26. His illness had confined him to his home for two months prior to his death. He leaves a wife and two daughters.



GEORGE A. O'ROURKE

Mr. O'Rourke was very active in boiler matters, having attended the meetings of the A.S.M.E. Boiler Code Committee regularly and having done a good deal of work in cooperation with that body. He was also, by virtue of his position as head of boiler-inspection work in the State of New York, a member of the National Board of Boiler and Pressure Vessel Inspectors and had participated in its organization at its first annual meeting at Detroit in February, 1921. His enthusiam in this kind of work has been one of his notable characteristics.

William H. Furman is now serving as acting chief engineer of the bureau in Mr. O'Rourke's place.

Primary Horsepower at Manufacturing Establishments*

The census of manufactures for twelve states, the District of Columbia and Porto Rico has been summarized so that the changes in power-plant installations of manufacturing establishments are now known, and the following table shows the extent to which there has been an increase or decrease in the primary horsepower. The change in the number of establishments and in the number of wage earners employed by them is also shown for the sake of comparison. From these data are also calculated the ratios of horsepower to man-power employed.

The total primary horsepower in the manufacturing establishments of the twelve states and the District of Columbia in 1914 was approximately 770,000 hp. In 1919 in the same territory there was 1,045,000 hp., or an increase of 35.5 per cent in five years. Seven of the states show a greater percentage increase in horsepower than in number of wage earners. Therefore in these localities the ratio of horsepower per man-power has increased by amounts varying from 4 to 100 per cent. In the same period, however, the other seven of the subdivisions thus far reported show a greater increase in number of wage earners than in horsepower and, therefore, a decrease in the ratio of horsepower per man-power. In this connection, however, it should be noted that most of the states reported are those where a relatively small number of establishments are found and in no case is any large industrial state included. It may be expected, therefore, that somewhat different conditions may prevail for the country as a whole when data are available for these larger and more important industrial districts.

A separate report has been made for fourteen cities in which comparison between 1914 and 1919 conditions is possible. In most cases these cities are within the states for which totals are available. They show that in five cases the horsepower per man-power has increased, but in nine cases it has decreased. On the average throughout the states and cities reported, the ratio of wage earners to primary horsepower has apparently changed little, though apparently a very slight increase in the ratio has occurred. In this connection one should note the fact that the horsepower here reported includes only the isolated-station power-plant capacity at the manufacturing establishments. It does not include the power or power-plant capacity of central stations or water-power development supplying manufacturing plants through public-utility lines. If this latter source of power were included it is probable that every subdivision of the country would show an increase in the power consumption, and in almost every case an increase in the ratio of power to wage earners.

In the fourteen cities referred to, eleven report increases in manufacturing-plant horsepower capacity, ranging from 18 to 200 per cent. Only three small cities, representing a total of only 5,000 hp. installed, show slight decreases in the plant horsepower installation. These three cities are all in Arizona and Nevada; they report only 7 to 17 per cent decrease. Here again this decrease represents isolated-station power-plant capacity and takes no account of the central-station or public-utility supplies, which may have increased in the meantime.

Only three large cities have thus far been reported; namely, Washington, D. C.; Wilmington, Del.; and Los Angeles, Cal. These three all show marked increase in horsepower installation, now having respectively 33,000, 51,000 and 95,000 hp. installed at the manufacturing plants reporting.

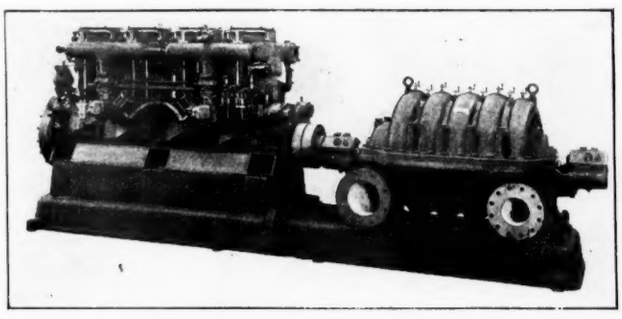
*Preliminary reports of census of manufactures—1919.

INDUSTRIAL POWER—PARTIAL REPORT OF CENSUS OF MANUFACTURES, 1919

State	Percentage Increase in No. of Establishments 1914 to 1919.	Percentage Increase in Wage Earners 1914 to 1919	Primary Horsepower (Thousands) 1914	1919	Percent Increase	Horsepower per Man-power 1914	1919	Percent Increase
Arizona	48	19	54.7	96.4	76	7.9	11.7	48
Colorado	24	29	162.8	194.6	20	6.0	5.5	-8
Dist. Columbia	15	19	24.7	33.0	34	2.8	3.1	11
Idaho	32	56	50.3	72.2	44	5.6	5.2	-7
Montana	38	25	91.7	169.1	84	6.7	9.9	48
Nebraska	16	45	90.2	125.8	40	3.6	3.4	-5
Nevada	-8	-15	18.7	13.9	-26	5.2	4.5	-13
New Mexico	5	52	3.8	5.7	52	7.7	10.0	30
North Dakota	28	36	15.0	17.8	18	4.6	4.0	13
South Dakota	58	69	16.3	22.4	37	4.3	3.5	18
Utah	5	36	59.5	92.2	55	4.3	4.9	14
Vermont	1	2	173.9	185.1	6	5.3	5.5	4
Wyoming	70	120	10.0	17.6	76	3.4	2.7	-20
Porto Rico	-34	3	34.0	67.2	98	2.1	4.2	100

Pumps at 350 Pounds Pressure

A five-stage pump has recently been built for the extremely high pressure of 350 lb. to the square inch (said to be the highest developed by any gasoline-engine-driven unit in the country). Its capacity is 900 gal. per min. at



AN UNUSUALLY HIGH PRESSURE PUMPING UNIT

1,200 r.p.m., and it is direct-connected to an eight-cylinder dual-valve sterling gasoline engine, which develops 242 b.hp. at 1,200 r.p.m. and 300 b.hp. at 1,500 r.p.m.

It was designed and built by the Lea Courtenay Co. to be installed at Ironwood, Mich., as a reserve unit pending completion of the water-works. It is to pump through a 20-in. main 4½ miles long.

Washington News

Progress in Coal Regulation

By Paul Wooton
Washington Correspondent

A law requiring compulsory returns to the government sufficient to make possible practical fact-finding that will give all concerned the information necessary to good business procedure seems assured at the present session of Congress. The Senate Committee on Interstate Commerce has authorized a report to the Senate urging the passage of the Frelinghuysen coal stabilization bill. This measure, formerly known as the coal commissioner bill, vests the Secretary of Commerce with the authority to require accurate returns as to coal production and consumption. The bill also provides for a comprehensive fact-finding program as to many phases of the coal business and directs the gathering of data on a number of the technological problems involved in the coal trade.

To determine just the types of information essential to the intelligent conduct of the coal business a conference is to be held at which representatives of producers, consumers, the railroads and the government will be present. Senator Frelinghuysen, the author of the bill, will be a party to the conference, as will the Secretaries of Commerce and of the Interior. If it should develop at this conference that changes should be made in the bill, as it has already been presented to the Senate, these changes will be embodied in committee amendments and the amendments can then be made a part of the bill when it is taken up for consideration in the Senate.

It is expected that the course of the bill will be expedited, since it is understood to be the Administration's policy to try out fact-finding before resorting to regulatory legislation. Many members of Congress believe that publicity of the essential facts involved will be all that is necessary to check abuses. At the same time, with reliable information available to producers and consumers alike, it is believed that additional stabilization of the industry can be secured.

Senator Frelinghuysen himself is very much in favor of the use of fact-finding as a corrective, rather than regulation. Just prior to reporting out the bill he gave expression to his opinion in the following language:

Expressing my own opinion, I think regulation will create more confusion and it will result in greater disorganization. Could not more be accomplished through co-operative effort than through regulation, arbitrarily established and temporary in character, which would disorganize the whole machinery of this industry?

J. D. A. Morrow, vice-president of the National Coal Association, appeared before the Frelinghuysen subcommittee on May 4. Among other things Mr. Morrow said:

It is the belief of the men who make up the executive committee of the National Coal Association that the current collection and publication of businesslike information as to some features of the coal industry, by a practical branch of the government, can be very helpful to the coal industry itself, as well as to the public.

If the facts had been available last summer and had been published widely, so that all producers and buyers could have been informed, it would have had a stabilizing effect on the situation. The individual consumer and the individual producer could have seen plainly that there was no warrant for $15 coal. We favor the collection of that type of information.

In the near future the Frelinghuysen subcommittee is expected to bring out a seasonal freight-rate bill. It is believed that this bill will authorize the Interstate Commerce Commission to reduce rates for seasonal periods to encourage a more nearly equal distribution of coal shipments throughout the year. Senator Frelinghuysen condemns the practice of buying coal on a hand-to-mouth basis. He says that it is unthinkable that a great industrial nation, using 500,000,000 tons of coal annually, should operate on such a basis and not carry reasonable surpluses, when unfavorable weather conditions and strikes are potential menaces to the regularity of supply.

New Publications

PYROMETRIC PRACTICE. By Paul D. Foote, C. O. Fairchild and T. R. Harrison. Published by the Department of Commerce, Washington, D. C., 1921. Paper; 7 x 10 in.; 326 pages; 185 illustrations. Price, 60 cents.

This is Technologic Paper No. 170 of the Bureau of Standards, the sort of volume which, if published commercially and bound in cloth, would sell at several times the price charged by the Government, and be worth every cent of it. In addition to a technical exposition of the scientific points involved, it is replete with information and suggestions for the users of high-temperature apparatus and processes in laboratory, shop and foundry. The chapter on Industrial Applications of Pyrometry describes its application in glass manufacture, portland cement kilns, ceramic industries, coke making, steel manufacture, etc., illustrations of which, from photographs, are contained in the appendix. Throughout the paper "important technical features of different instruments are considered with a view to making available to the users of such instruments information that is essential to determining the adaptability to the various applications of high-temperature measurement, as well as to secure the most efficient use of the instruments when installed." The book is obtainable from the Superintendent of Documents, Government Printing Office, Washington, D. C., on receipt of the purchase price.

Personals

William H. Yates, consulting engineer, has removed his offices from 111 Broadway to 16 East 41st St., New York City.

John J. Swan has become associated with the Engineering Business Exchange of New York City and has moved there from Indianapolis. He has been with the Prest-O-Lite Company for the last two years.

J. H. Libby is now associated with H. M. Haven and William W. Crosby, engineers and architects, Boston, Mass. He recently resigned as engineer of power and electric lines of the Eastern Massachusetts Street Railway Co.

W. S. Lee, since 1905 vice president and chief engineer of the Southern Power Co., Charlotte, N. C., and prominently identified with water-power development and electrification of industries in the South, has been elected president of the Piedmont & Northern Railway.

Society Affairs

The National Electric Light Association is to hold its annual convention May 31-June 3 at the new Drake Hotel, Chicago. Further particulars will be found elsewhere in this issue.

Metropolitan Section, American Welding Society, is to hold a "Practical welder's night," May 17, at which one of three talks on welding will be "Welding of Oil-Engine Cylinders," by A. F. Keogh, of the Sound Welding Co.

The Engineer's Society of Western Pennsylvania will meet May 17, at the William Penn Hotel, Pittsburgh, to hear a "Description of the Manufacture of Elevating, Conveying and Power Transmission Chains," by G. L. Morehead, manager of the Link-Belt Co.

The American Society of Safety Engineers will meet at the Engineering Societies Building, New York City, May 27, to consider three papers: "High Tension Electric Generation and Transmission," "Electrical Hazards—Terminal Equipment of the New York Central R.R. Entering New York City" and "Small Motor Problems and Industrial Control."

The Franklin Institute will meet May 18, at 3:30 p.m., in Philadelphia, for the presentation of honorary membership to General Pershing and of the Franklin Medal and honorary membership to M. Jusserand, the French Ambassador, and to Frank J. Sprague. Charles Fabry will speak on "Light Radiation" and Mr. Sprague on "Electric Traction."

Central Pennsylvania Section, A. S. M. E., will hold its first meeting at 10 a.m. May 21, at the University Club, State College, Pa. The program includes: An inspection of the special engineering features of the Pennsylvania State College; an address by Fred Miller, past president A. S. M. E.; lunch at the University Club; and addresses by Dean R. L. Sackett and by Dr. Hollis, past president, A. S. M. E. The general subject spoken on will be "Present Day Duties and Opportunities Before Engineers." The meeting will be over at 3 p.m., when there will be an inter-college baseball game.

Business Items

C. H. Wheeler Manufacturing Co. has moved its New York office from 114 Liberty St. to 50 Church St.

Dean, Fairbank, Obrieght and Hirsch, patent, trade mark and corporation lawyers, have moved to 15 Park Row, New York City.

The Lunkenheimer Company has moved its Chicago branch store from 188 North Dearborn St. to 568 West Washington Boulevard.

Murrie & Co., Inc., 74 Broadway, New York City, has placed Edward R. Knowles in charge of their pulverized coal and gas fuel engineering work.

Manning, Maxwell & Moore, Inc., 119 West 40th St., New York City, has announced the election of J. M. Davis as its president to succeed the late A. J. Babcock.

The Burt Mfg. Co., Akron, O., has announced that J. Asa Palmer, general manager of the company for a number of years, has been elected president and general manager in place of the late W. F. Warden.

The McClave-Brooks Co., Scranton, Pa., manufacturer of McClave stokers, grates and steam blowers, has moved its New York office from 50 Church St. to 1502 Printing Crafts Building, 461 Eighth Avenue.

The American Engineering Co., of Philadelphia, has appointed Isaac Hardeman as sales agent representing the Taylor Stoker in the states of North Carolina, South Carolina and Georgia. Mr. Hardeman's office is at 816 Realty Building, Charlotte, N. C.

FUEL PRICES

BITUMINOUS COAL

The following table shows the trend of the spot steam market in various coals (mine-run basis, f.o.b. mines):

Coal	Market Quoted	May 3 1921	May 10 1921
Pocahontas,	Columbus	$3.35	$3.50
Clearfield,	Boston	2.35	2.00@2.65
Somerset,	Boston	2.95	2.25@3.60
Pittsburgh,	Pittsburgh	2.25	1.90@2.25
Kanawha,	Columbus	2.00	2.25
Hocking,	Columbus	2.00	2.10@2.25
Pittsburgh No.8	Cleveland	2.20	2.15@2.25
Franklin, Ill.,	Chicago	3.40	3.25@3.50
Central, Ill.,	Chicago	2.95	2.75@3.15
Ind. 4th vein,	Chicago	3.25	3.00@3.75
Standard,	St. Louis	1.80	1.90
West Ky.,	Louisville	2.10	2.00
Big Seam,	Birmingham	2.95	2.85@3.10
S. E. Ky.,	Louisville	2.60	2.50@2.75

DIESEL FUEL OIL

New York — On May 12, Port Arthur light oil, 23@25 deg. Baumé, 5¾c. per gal. 30@35 deg., 6½c. per gal. f.o.b. Bayonne, N. J.

Chicago — May 7, for 24@28 deg. Baumé, 60@70c. per bbl.; 32@34 deg. 2¼@2½c. per gal. In tank cars f.o.b. Oklahoma refinery, or freight adjusted.

St. Louis—May 7, prices f.o.b. cars, tank lots: 24@26 deg. Baumé, 65c. per bbl.; 26@28 deg., 70c.; 28@30 deg., 80c.; 32@34 deg., 3c. per gal.

Baltimore—On April 23 f.o.b. refinery; in tank-car lots; 24+ deg. Baumé, 6½c.; 26@28 deg., 6½c.; 30@34 deg. 7¾c.; 14+ deg. $1.95 per bbl.; 16@20 deg., $1.85 per bbl.

Pittsburgh—On May 2, f.o.b. refinery; Pennsylvania, 36@40 deg., 5c. per gal.; Oklahoma, 24@30 deg., 65c. 32@34 deg., 75c. per bbl.; gas oil, 32@34 deg. 2½c. per bbl., 36@38 deg., 2¾c. 38@40 deg. 3c.

Philadelphia—On April 12; 26@28 deg. Baumé, Oklahoma, 85c. per bbl.; 30 @ 34 deg., Oklahoma (group 3) 3c. per gal.; 16@20 deg., Seaboard, 5c. per gal.

Cincinnati — April 23, for 22@28 deg. Baumé, 7.34c. per gal.

Cleveland — April 23, for 22@28 deg. Baumé, 5½c. per gal.

New Construction

PROPOSED WORK

Me., Winthrop—C. M. Bailey Sons Oilcloth Co. is having plans prepared for a 3 story, 50 x 145 ft., 3 story, 95 x 120 ft. and a 1 story, 80 x 98 ft. factory buildings. About $250,000. The Austin Co., 217 Bway, New York City, Engr.

Mass., Northampton—J. W. Ames, Archt., 15 Exchange St., Boston, will receive bids for three dormitories each 3 story, 40 x 200 ft., for Smith College here. About $500,000.

Mass., Springfield—Cordner & Montague, 373 Bay St., plans to build a 3 story, 160 x 210 ft. garage and warehouse near Railroad St. About $350,000. F. L. Knowlton, Engr.

Conn., Ansonia—The Seccombe Bros., 185 Howard Ave., is in the market for one No. 1 Schramm portable compressor.

Conn., New Britain—The Bd. Educ. plans to build a 3 story grade school on Washington St. About $370,000.

Conn., New Haven—James D. Kauffman, 42 Church St., is having plans prepared for a 2 story, 205 x 210 ft. apartment house on Prospect and Highland Sts. About $400,000. J. Weinstein, 6 Church St., Archt.

Conn., West Hartford — The Kingswood School, 351 Farmington Ave., Hartford, plans to build 4 school buildings including an administration, gymnasium, refectory and servants buildings on Outlook Ave. About $600,000. E. Sherrill Dodge, 15 Exchange St., Boston, Mass., Archts.

N. Y., Central Islip—The State Hospital Comn., Capitol, Albany, will receive bids June 1 for construction work-expansion and anchoring chambers for underground connections and addition to power house, heating work- underground service connections, additional power house equipment, etc.

N. Y., Central Islip—The State Hospital Comn., Capitol, Albany, received bids for a refrigerating unit for kitchen at the State Hospital here, from Richmond & Daniels Inc., 90 West St., New York City, $4,080; Automatic Refrigerating Co., 1618 Capitol Ave., Hartford, Conn., $4,160; Shipley Constr. & Supply Co., 659 Bway, Albany, $4,315.

N. Y., Long Island City—The New York Telephone Co., 15 Dey St., plans to build a telephone exchange building on 9th St. through to 10th St., near Van Alst Ave. About $850,000.

N. Y., Middletown—The State Hospital Comn., Capitol Albany, received bids for the installation of electric motors for the laundry machinery at the state hospital here, from T. F. Jackson, 99 John St., New York City, $4,590; E. W. Tompkins Co., Albany, $4,943 Jandous Electric Equipment Co., 109 West 31st St., New York City, $5,610.

N. Y., New York—(Brooklyn Boro) The State Hwy. Dept., 55 Lancaster St., Capitol Albany, will receive bids May 19 for installing a heating system in the Barge Canal Terminal headhouse at Gowanus Bay here. About $5,000.

N. Y., New York — (Manhattan Boro) The Buyers' & Merchants Exch., Woolworth Bldg., is having plans prepared for about a 25 story 160 x 175 ft. office building on Bway. between 36th and 37th St. About $3,500,000. H. Babcock, Secy.

N. Y., New York — (Manhattan Boro) Lebanon Hospital, Westchester Ave., will soon receive bids for an addition to hospital. About $500,000. C. B. Meyers, Union Sq., Archt. and Engr.

N. Y., New York — (Manhattan Boro) The Neesmont Realty Corp., c/o Victor C. Farrar, Archt. and Engr., 4 East 39th St., is having plans prepared for an 11 story, 60 x 100 ft. office building at 49 West 45th St. About $400,000.

N. Y., New York — (Manhattan Boro) The New York Edison Co., Irving Pl. and 15th St., will soon award the contract for a sub-station on 40th St. and 1st Ave. Cost about $200,000. W. Whitehill, 12 Elm St., Archt. and Engr.

N. Y., New York — (Manhattan Boro) The Society of Professional Automobile Engrs., Bway and 67th St., is having revised plans prepared for a 5 story, 75 x 100 ft. club house including a steam heating system at 153 West 64th St. About $300,000. W. Haefeli, 229 West 42d St., Archt. and Engr.

N. Y., Wende — (Millgrove P. O.) Erie Co., City Hall, Buffalo, is having plans prepared for a 2 story infirmary here. About $500,000. C. E. Speich, 575 Ellicott Sq. Bldg., Buffalo, Archt.

N. J., Bridgeton — R. McAllister representing owner plans to build a 6 story hotel including a steam heating system. About $300,000. Dreher, Churchman, Paul & Ford, 1424 Walnut St., Philadelphia, Pa., and W. L. Custer, Camden, N. J., Archts.

N. J., Glassboro—The State Bd. Educ., State House, Trenton, engaged Arnold & Moses, Merchantville, N. J., and Guilbert & Betelli, 546 Broad St., Newark, to prepare plans for a normal school here. About $350,000.

N. J., Trenton — The Bd. Educ., State House, plans to build a school for the deaf. About $300,000.

Pa., Dormont—(Pittsburgh P. O.) The School Bd. will receive bids May 19 for a 2 story, 75 x 194 ft. school. About $275,000. Press C. Dowler, Magee Bldg., Pittsburgh, Archt. and Engr.

Pa., Erie—The Columbia Amusement Co., c/o Arland W. Johnson, Archt. and Engr., 469 5th Ave., New York City, plans to build a 10 story theatre and office building on State St., here. Estimated cost, $750,000.

Pa., McKeesport — McKeesport Hospital will receive bids after June 1 on the superstructure of a 3 and 4 story hospital building including nurses' home, power building, etc. About $250,000. S. F. Heckert, Bessemer Bldg., Pittsburgh, Archt.

Pa., Philadelphia — The Zion Hosiery Mills, 265 South 4th St., is in the market for one hp., 220 volt, d.c. motor.

Pa., Pittsburgh — W. C. Davison, Gas Burner & Welding Co., 3145 Penn Ave., is in the market for a small 100 hp. boiler. F. F. Davison, Mgr.

Md., Baltimore — The Boumi Temple, Nobles of the Mystic Shrine, c/o Otto G. Simonson, Archt., Maryland Casualty Tower Bldg., is having plans prepared for a 6 story, 110 x 200 ft. mosque and theatre on Howard St. About $1,100,000.

W. Va., Bluefield—The Cole Realty Co. is having plans prepared for a 12 story, 60 x 175 ft. hotel at the corner of Federal and Scott Sts. About $800,000. A. B. Mahood, Archt.

W. Va., Fairmont—C. E. Hutchinson is having plans prepared for a 4 story business building on Washington St. About $500,000. Jones & Nuzum, Archts.

W. Va., Maidsville—(R. D. Vanvoohis) The North American Coal Co. plans to reconstruct power house. About $10,000.

S. C., Camden—The Bd. of Trustees of the Camden School Dist., will receive bids June 1 for a school and auditorium building.

Fla., Leesburg—The city plans to vote either $75,000 bonds to enlarge power plant or $40,000 bonds to build a power line to Inverness where current will be furnished.

O., Cleveland—The Cleveland Columbian Bldg. Co., 3422 Euclid Ave., plans to build a 3 and 6 story office and club house including a steam heating system. About $350,000. F. Stilman Fish, 6508 Euclid Ave., Archt.

Ind., Ft. Wayne—The Bd. Educ. plans to build a 3 story, 211 x 230 ft. high school including boilers, etc. About $500,000. Griffith & Goodrich, 211 East Berry St., Archts.

Ind., Ft. Wayne—The S. J. Gregory Theatre Co., c/o J. Pappas, plans to build a 3 or 4 story theatre. About $500,000.

Ind., Frankfort—The Bd. of Pub. Wks. plans to build a 1 story, 100 x 100 ft. power plant. About $50,000. J. D. Lyon, Union Central Bldg., Cincinnati, O., Engr.

Ind., Newcastle—The Indiana Village of Epileptics plans to build 12 buildings including 1 story power plant, laundry, etc. in addition to asylum here. About $500,000. Hilbert Foltz, 843 Lenicke Annex, Indianapolis, Archt.

Ind., South Bend — The La Salle Hotel Corp. plans to build an 8 story hotel. About $750,000.

Ill., Chicago—The Ascher Bros. Theatrical Enterprises, 220 South State St., will soon award the contract for a 2 story, 80 x 120 ft. theatre and store building including a steam heating system on Claremont Ave. and 63d St. About $325,000. C. Howard Crane, 127 North Dearborn St., Archt.

Ill., Chicago—G. W. Cook & Co., c/o R. F. France, Archt., 155 North Clark St., plans to build a 5 story, 140 x 140 ft. apartment hotel including a steam heating system on Washington Blvd. and Oak Park Ave. About $500,000.

Wis., New Holstein—The Municipal Electric Power Comn., c/o H. V. Edens, Chn., will receive bids until May 30 for an extension of 66,000 volt transmission line. About $50,000.

Wis., Oshkosh—The Oshkosh Tractor Co. plans to build a 5 story, 60 x 210 ft. tractor factory on Main St. About $200,000. Architect not yet appointed.

Wis., Waukesha—I. J. Frisch, 5433 Prairie St., Chicago, Ill., representing a Stock company, is having plans prepared for a 3 story, 150 x 255 ft. sanitarium here. About $325,000. Z. E. Smith, 305 East 55th St., Chicago, Ill., Archt.

Minn., Ellsworth—The city voted $30,000 bonds to improve and rebuild the electric light system. The plant will be changed from d.c. to a.c. E. A. Meeter, City Clk. Druer & Milinowski, 512 Globe Bldg., St. Paul, Engrs.

Minn., Minneapolis — The Bd. Educ. is having plans prepared for a 2 story junior high school to be known as Bryant High School. A steam heating system will be installed in same. About $400,000. G. F. Womrath, Business Supt. E. H. Enger, City Hall, Engr.

Minn., Minneapolis—The Bd. Educ. will receive bids May 25 for a 2 and 3 story, 325 x 580 ft. high school to be known as the Jordan Junior High School on 29th Ave. North and James Ave. A steam heating system will be installed in same. About $600,000. G. F. Womrath, Business Supt. E. H. Enger, 305 City Hall, Engr.

S. D., Brookings—The city voted $60,000 bonds to remodel and build addition to light and power plant.

S. D., Ft. Pierre—The City, c/o Rex Terry, Secy. of Water Comn., will soon receive bids for one 100 hp. pumping engine, one centrifugal electric pump, one 75 kva. generator for water plant. About $25,000. Cory & LeCocq, Aberdeen, Engrs.

S. D., Mitchell—St. Joseph's Hospital will soon receive bids for a 5 story, 66 x 155 ft. hospital including a steam heating and refrigerating plant. About $300,000. Sisters of Presentation Order are in Charge. E. H. Lundie, 530 Endicott Bldg., St. Paul, Minn., Engr.

S. D., Watertown — The city plans to build a waterworks system and power house. About $75,000.

S. D., Watertown—The city will receive bids May 23 for additional power plant equipment. A. J. Mueller, Supt.

Mo., Jasper — The city is having plans prepared for a waterworks and electric light plant. About $40,000. James Hull, Mayor. A. C. Moore, Independence Bldg., Joplin, Engr.

Mo., St. Louis—The Hotel Realty Co., 721 Locust St., is having plans prepared for a 13 story, 113 x 165 ft. addition to the Jefferson hotel on 13th and Locust Sts. About $2,500,000. F. J. Wade, Pres. T. P. Barnett Co., Arcade Bldg., Archt.

Mo., St. Louis—C. Ulman, 4540 Lindell Ave., is having plans prepared for two 8 story, 50 x 100 ft. and 95 x 98 ft. apartment houses including steam heating system on Kingshighway and Lindell Ave. About $500,000 each. P. J. Bradshaw, International Life Bldg., Archt.

Tex., Brownsville—Mrs. C. Moore is having plans prepared for a 6 story hotel including a steam heating system. About $250,000. A. B. Ayres, Bedell Bldg., San Antonio, Archt.

Okla., Sayre—The town has engaged V. V. Long & Co., Colcord Bldg., Oklahoma City, to prepare plans and submit estimates to build an electric distribution and power plant, here. About $48,000.

Col., Denver—The School Bd. will receive bids until June 1. for a 2 story, 240 x 300 ft. North Side or Woodbury Junior high school including a steam heating system on West 40th and King Sts. About $600,000. H. W. Edwards, 1172 Stout St., Archt.

Utah, Duchesne—The Great Basin Power Co., Salt Lake City. plans to build a hydroelectric plant to have a capacity of 10,000 hp. here About $1,500,000. J. Lipman, Pres. H. A. Strauss, Salt Lake City, Engr.

Cal., Wilmington — The Union Oil Co., Union Oil Bldg., Los Angeles, is having plans prepared for a 3 story, 110 x 144 ft. administration building including a steam heating system here. About $305,000. R. J. Reed, Union Oil Bldg., Los Angeles, Engr. J. T. Howard, 1st Natl. Bank Bldg., San Francisco, Archt.

Ont., Hanover—The city has engaged F. W. Thoroed, Engr., 167 Avenue Rd., Toronto, to prepare plans and submit estimates for additions to the waterworks system including pumps, reservoir, etc. About $50,000.

Ont., Toronto — James, Proctor & Redfern, Engrs., 36 Toronto St., will receive bids May 19 for a pump house and waterworks reservoir to include electric centrifugal pumps, etc. About $150,000.

CONTRACTS AWARDED

Vt., Montpelier—The Natl. Life Insurance Co. has awarded the contract for an 8 story office building including a steam heating system on State St., to L. D. Willicutt & Sons, Inc., 146 Summer St., Boston. About $1,000,000.

Mass., Boston—Reynolds Estate, c/o Parker, Thomas & Rice, Archts., 110 State St., has awarded the contract for an 8 story, 34 x 100 ft. office building on Winter St., to L. D. Willicutt & Son Co., 146 Summer St.

Mass., Fall River—Pilgrim Hall, c/o McKim, Mead & White, Archts. and Engrs., 101 Park Ave., New York City, has awarded the contract for an addition to hall, to Roy H. Beattie, Buffington Bldg. About $500,000.

Mass., Newtonville—(Boston P. O.)—The Security Mills Inc., West Newton, has awarded the contract for a textile mill including a 1 story, 75 x 250 ft. mill building, 1 story, 55 x 90 ft. dye house and a 1 story, 30 x 35 ft. boiler house, to W. T. Read Co., 201 Devon St., Boston. About $125,000.

Mass., Reading—The Commonwealth of Massachusetts, 1 Ashburton Pl., Boston, will build a 1 story, 30 x 40 ft. sewage disposal pumping station here. About $35,000. Work will be done by day labor. Noted April 26.

N. Y., Buffalo—The University of Buffalo, High and Main Sts., has awarded the contract for a chemistry building on Main St. and the City Line, to J. W. Cowper Co., Fidelity Bldg., at $540,000.

N. Y., New York — (Bronx Boro) The Bd. Educ., 500 Park Ave., New York City, has awarded the contract for installing a heating and ventilating system in the annex of P. S. 43 on 136th St. and Brown Pl., to the Raisler Heating Co., 129 Amsterdam Ave., New York City, at $41,500. Noted May 10.

N. Y., New York—(Brooklyn Boro) The Newkirk Plaza Bldg. Corp., c/o Seelig & Finkelstein, Archts. and Engrs., 44 Court St., will build a 5 story, 60 x 120 ft. apartment house on Newkirk Ave. About $300,000. Work will be done by day labor.

N. Y., New York (Manhattan Boro)—C. Carroll Inc., 213 West 42nd St., will build a theatre, store and office building on 7th Ave. and 50th St. About $500,000. Work will be done by day labor. Noted April 19.

N. Y., New York — (Manhattan Boro) The Columbus Circle Arcade Co., c/o Francisco & Jacobus, Archts. and Engrs., 511 5th Ave., has awarded the contract for a 23 story top addition to present office building on Bway, 8th Ave. and 57th St., to the Thompson Starrett Co., 49 Wall St. About $3,000,000.

N. Y., New York — (Manhattan Boro) J. Cracky, 64 6th Ave., has awarded the contract for a 4 story garage on University Pl. and 12th St., to C. R. Hedden, 763 Broad St., Newark, N. J. About $250,000.

N. J., Jersey City—The Trust Co. of New Jersey, Hudson St., has awarded the contract for an 11 story bank and office building on Bergen and Sipp Aves., to Hegeman & Harris Co., 185 Madison Ave., New York City. About $1,000,000.

Pa., Ardmore — The Main Line Amusement Co. Inc., has awarded the contract for a 2 story, 80 x 200 ft theatre, store and office building including a steam heating system, to Stuckert & Co., Crozer Bldg., Philadelphia, at $300,000

Md., Baltimore—The Campagnia Apartment Co., c/o Clyde & Nelson Friz, Archts., 2010 Lexington Bldg., has awarded the contract for a 4 story apartment hotel including heating and ventilating system on 40th and Oak Sts., to the Price Constr. Co., Maryland Trust Bldg., at $400,000.

Md., Baltimore—Fulton E. Yewell, 3600 Forest Park Ave., will build a 3 story, 80 x 165 ft. apartment house including a steam heating system on Hilton Ave. and Liberty Heights. About $300,000. Work will be done by day labor.

O., Cleveland — The German Lutheran Hospital, 2609 Franklin Ave., has awarded the contract for a 4 story, 117 x 140 ft hospital at 2609 Franklin Ave., to the Emerson Co., 1900 Euclid Ave. About $250,000.

O., Columbus—The Federal Ice & Refrigerating Co., 20th and Whitcomb Sts., has awarded the contract for a 1 and 2 story ice plant on 20th St., to Blome, Surek Co., 139 North Clark St., Chicago, Ill., at $90,000.

O., Cuyahoga Falls—The Bd. Educ. has awarded the contract for installing a heating system in the proposed 2 story, 115 x 136 ft. school, to W. J. Scholl Co., 109 East Commerce St., Youngstown, at $44,000.

Ind., Gary—The Barnes Ice & Coal Co., 105 Bway, will build a 2 story, 72 x 100 ft. ice plant. About $75,000. Work will be done by day labor.

Ind., Huntington—J. F. Bippus has awarded the contract for a 4 story, 115 x 150 ft. hotel, to Stevens & Co. About $300,000.

Ill., Herrin—The Herrin Hotel Co., Fletcher Savings Bank Bldg., Indianapolis, has awarded the contract for a 3 story, 109 x 136 ft. hotel including a steam heating system on East Madison and 14th Sts., here, to the Hoffman Constr. Co., Indianapolis. About $250,000.

Wis., Kenosha—The Wisconsin Theatre Co., 172 Market St., has awarded the contract for a 3 story, 80 x 150 ft. theatre and store building including a steam heating system on Chicago St., to H. W. Horst, Brumder Bldg., Milwaukee. About $275,000. Noted Feb. 15.

Wis., Milwaukee—The Harley Investment Co., 1388 Humboldt Ave., has awarded the contract for a 6 story, 132 x 186 ft. apartment hotel on Grand Ave., to Riesen Bros., 1388 Humboldt Ave. About $700,000. Noted May 17.

Wis., Milwaukee — The Milwaukee Electric Ry. & Light Co., 2nd and Sycamore Sts. will build a 2 story, 30 x 143 ft. substation on Clinton St. About $35,000. Work will be done by day labor.

Ia., Cedar Rapids—The Bd. Educ., 705 1st Ave., has awarded the contract for a 3 story, 108 x 121 ft. school including a steam heating system, to T. Stark & Co., 118 59th St. About $500,000.

Ia., Ft. Madison—The Hinde Dauche Co. has awarded the contract for a 2 and 3 story, 100 x 340 ft. paper mill including a steam heating system, to H. K. Ferguson, 6523 Euclid Ave., Cleveland. About $1,000,000.

Ia., Ottumwa — The Bd. Educ. has awarded the general contract for a 3 story, 192 x 275 ft. high school and a 1 story, 60 x 60 ft. power plant, to the Natl. Contg. Co., 311 Metropolitan Bank Bldg., Minneapolis, at $412,888. Noted May 3.

Col., Denver — The Jewish Consumptive Relief Society, 5801 West Colfax Ave., has awarded the contract for a 3 story, 50 x 400 ft. sanitarium and two 2 story, 30 x 125 ft. pavilions, to F. J. Kirchoff Constr. Co., at $250,000.

Utah, Curtis—The Salt Lake & Utah R.R. Co., 10 S and 1 W., has awarded the contract for a 500 kw. sub-station, to the Westinghouse Electric Co., Walker Bank Bldg., Salt Lake City, at $50,000.

Cal., Long Beach — W. W. Paden, 427 Story Bldg., Los Angeles, has awarded the contract for an 8 story, 100 x 130 ft. hotel including a steam heating system on Ocean Blvd., to the Foundation Co., Higgin Bldg., Los Angeles, at $82,500.

Que., Montreal—The Dupuis Freves Ltd., 447 Catharine St. East, has awarded the contract for a 5 story, 125 x 200 ft. departmental store to the Atlas Constr. Co., 37 Belmont St. About $500,000.

The SEARCHLIGHT Advertising in This Paper

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